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USER'S MANUAL FOR STEADY STATE AND TRANSIENT THERMAL ANALYSIS OF A SHAFT-BEARING SYSTEM (SHABERTH)

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Predicting the performance of helicopter engine and transmission bearings following loss of normal circulating lubrication is an important part of an aircraft vulnerability estimate. The program SHABERTH was developed to provide the means of estimating the mechanical and thermal state of critical components within a shaft-bearing-housing drive system, first in the normal service (steady state) lubricated condition, and second, in the oil-starved (transient) condition. The program is installed on the BRL Univac 1108 system and is used

20. ABSTRACT (Continued)

primarily to predict time-to-failure and failure mode after oil starvation; however, it will at the same time predict thermal dams, critical clearances and other significant behavioral features of the system under treatment, for both normal and dry operation. The program is being used for problems concerning domestic developmental aircraft and non-domestic aircraft, for which no physical test data or hardware are available.

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I. INTRODUCTION

This manual describes the fourth generation of SHABERTH, "A Computer Program for the Steady State and Transient Analyses of Shaft Bearing Systems Including Ball Cylindrical and Tapered Roller Bearings." The original program, AE72Y003, was developed by Kellstrom (1) under U.S. Army Contract DAAD05-73-C-0011, sponsored by the Ballistics Research Laboratory at Aberdeen Proving Grounds. The present version was developed under U.S. Army Contract DAAD05-75-C-0747, also sponsored by the Ballistics Research Laboratories.

The master program consists of the following major sub-programs:

1) Bearing Analysis. These subprograms are largely based upon the methods of Harris, (2,3).

2) Three Demensional Shaft Deflection Analysis developed by Norlander and Friedrichson.

3) Bearing Dimensional Change Analysis based on the methods of Timoshenko, (4), and adapted to the shaft-bearing-housing system by Crecelius, (5).

4) Generalized Steady State and Transient Temperature Mapping and Heat Dissipation Analyses based on the methods of Harris, (6), Fernlund, (7) and Andreason, (8).

Although the primary function of all four generations of the program is to predict general bearing performance characteristics, and the bulk of the coding reflects this emphasis, the steady state and transient heat dissipation and temperature mapping sub-program may be used on a stand alone basis to model the thermal behavior of any system which can be represented by discrete temperature nodes.

The differences between the successive generations of the program reflect the development and installation of improved bearing lubrication and friction models, improved analysis of the bearing cage and improvements in the program structure which increased the program versatility and solution procedures. In the fourth generation, program capabilities were expanded to include tapered and cylindrical roller bearings with flanges.

The first generation of the program used the Newtonian lubricated friction models developed by Harris (2,3).

The second generation of the program which carries the designation AT74Y001 was created by Crecelius, Liu and Chiu under Air Force Contract No. F33615-72-C-1467 and Navy MIPR No. M52376-3-000007 and is documented by McCool, et al (9)*. In this effort, with Program AE72Y003 as the basis, the ball bearing subprogram was modified to include new models as follows:

1) An EHD film thickness model that accounts for i) thermal heating in the contact inlet using a regression fit to results obtained by Cheng (10) and ii) lubricant film starvation using theoretical results derived by Chiu (11).

2) A new semi-empirical model for fluid traction in an EHD contact (9), is combined with an asperity load sharing model developed by Tallian (12) to yield a model for traction in concentrated contacts which reflects various conditions of lubrication. Dry, partial and full EHD regimes are considered.

3) A model for the hydrodynamic rolling and shear forces in the inlet zone of lubricated contacts accounting for the degree of lubricant film starvation, (9).

4) Normal and friction forces between a ball and a cage pocket are modelled in a way that accounts for the transition between the hydrodynamic and elasto-hydrodynamic regimes of lubrication (9).

5) A model for the effect on fatigue life of the ratio of the EHD plateau film thickness to the composite surface roughness, (9).

Additionally, models for temperature viscosity and pressure viscosity variation as functions of temperature given by Walther (13) and Fresco (14) respectively, were adopted.

Program AT74Y001 is capable of analyzing only a single axially loaded ball bearing. The program cannot be used to analyze a multi-bearing system. All other capabilities are present however.

*Due to the similarities between major segments of SHABERTH AT77Y001 & AT74Y001, many sections of (9) have been included in this text, without modification.

The basis for the third version AT75Y004, was AT74Y001. The latent capability for the analysis of up to five ball and roller bearings subjected to general, (5 degrees of freedom) loading, was utilized. The models added to AT74Y001 were used in the calculation of both the ball and roller bearing friction forces and frictional heat generation rates. The third version also includes a model for the hydrodynamic rolling and slip forces in the inlet zone of lubricated line contacts, based on the work of Chiu, Ref. (15). Additionally, a cage model developed under NASA Contract No. NAS3-19739 Ref. (15) was added which allows the cage to move with up to three degrees of freedom versus the one degree of freedom permitted in Program AT74Y001. This cage model is used in the analysis of both ball and roller bearings.

This fourth generation program, SHABERTH, AT77Y001, differs from the preceding versions in that it can accomodate tapered roller bearings as well as ball and cylindrical roller bearings. Additionally, the capability of the cylindrical roller bearing analysis has been expanded to permit load to be carried by inner and outer ring flanges.

SHABERTH is intended to be as general as possible with the following limits on system size.

Number of bearings supporting the shaft - five (5) maximum

Number of rolling elements per bearing - thirty (30) maximum

Number of temperature nodes used to describe the system-one hundred (100) maximum

The program structure is modular and has been designed to permit substitution of new mathematical models and refinements to the existing models as the needs and opportunities develop.

2. PROBLEM FORMULATION AND SOLUTION

The purpose of the program is to provide a tool with which the shaft bearing system performance characteristics can be determined as functions of system temperatures. These system temperatures may be a function of steady state operation or a function of time variant conditions brought on by a change in the system steady state condition. Such a change would be the termination of lubricant supply to the bearings and other lubricated mechanical elements.

The program is structured with four nested, calculation schemes as follows:

1. Thermal, steady state or transient temperature calculations which predict system temperatures at a given operating state.

2. Bearing dimensional equilibrium which uses the bearing temperatures predicted by the temperature mapping subprograms and the rolling element raceway load distribution, predicted by the bearing subprograms, to calculate bearing diametral clearance at a given operating state.

3. Shaft-bearing system load equilibrium which calculates bearing inner ring positions relative to the respective outer rings such that the external loading applied to the shaft is equalized by the rolling element loads which develop at each bearing inner ring at a given state.

4. Bearing rolling element and cage load equilibrium which calculates the rolling element and cage equilibrium positions and rotational speeds based upon the relative inner-outer ring positions, inertia effects and friction conditions, which if lubricated, are temperature dependent.

The above program structure allows complete mathematical simulation of the real physical system. The program has been coded to allow various levels of program execution which prove useful and economical in bearing design studies.

These levels of execution are explained fully in Sections 3, 4, and 5.

The structure of the program and the nesting of the solution loops noted above can be seen clearly in the Program Flow Chart which is discussed in Appendix A.

The sections below present the systems of field equations which are solved in each of the nested calculation schemes. A more detailed discussion is contained in (1,9 and 15).

2.1 Temperature Calculations

Subsequent to each calculation of bearing generated heat rates, either the steady state or transient temperature mapping solution scheme may be executed. This set of sequential calculations is terminated as follows:

1. For the steady state case; when each system temperature is within EPI °Centrigrade of its previously predicted value, EPI is specified by the user. If it is zero or left blank, a default value of 1^{o} Centigrade is used. This criteria implies that the steady state equilibrium conditions has been reached.

2. The transient calculation terminates when the user specified time up is reached or when one of the system temperatures exceeds $600^{o}C$.

2.1.1 Steady State Temperature Map

The mechanical structure to be analyzed is thought of as divided into a number of elements or nodes, each represented by a temperature. The net heat flow to node i from the surrounding nodes j, plus the heat generated at node i, must numerically equal zero. This is true for each node i, i going from 1 to n, n being the number of unknown temperatures.

After each calculation of bearing generated heat, which results from a solution of the shaft-bearing system portion of the program, a set of system temperatures is determined which satisfy the system of equations:

$$q_i = q_{oi} + q_{gi} = 0 \text{ for all temperature nodes } i \qquad (2.1)$$

where q_{oi} is the heat flow from all neighboring nodes to node i

q_{gi} is the heat generated at node i. These values may be input or calculated by the shaft bearing program as bearing frictional heat

This scheme is solved with a modified Newton-Raphson method which successfully terminates when either of two condition are met:

$$\frac{\Delta t_i}{t_i} \leq EP2 \text{ for all nodes } i \tag{2.2}$$

where: Δt represents the Newton-Raphson correction to the temperature t at a given iteration such that, $t_{N+1} = t_N + \Delta t$ and $N + 1$, and N, refer to the next and current iteration respectively.

EP2 is a user specified constant. If EP2 is left blank or set to zero (0) a default value of 0.001 is used.

A second convergence criterion dependent upon EP2 is also used. In the system of equations, $q_{oi} + q_{gi} = 0$ for all nodes i, absolute convergence would be obtained if the right hand side (EQ) in fact reduced to zero (0). Usually a small residue remains at each node, such that $(q_{oi} + q_{gi}) = (EQ)i$.

The second convergence criterion is satisfied if:

$$\sum_{i}^{n} \left[\frac{(EQ)_i^2}{n}\right]^{\frac{1}{2}} \leq 100 \times EP2 \tag{2.3}$$

where n = number of equations in bearing solution

2.1.2 Transient Temperatures

In the transient case the net heat q_i transferred to a node i heats the element. It is thus necessary for heat balance at node i that the following equations are satisfied.

$$\rho_i \, C_{pi} \, V_i \, \frac{dt_i}{dT_i} = q_i \tag{2.4}$$

where ρ = density
Cp = specific heat
V = volume of the element
t = temperature
T = time

The temperatures, t_{oi}, at the time of initiation $T = T_s$ are assumed to be known, that is

$$t_i(T_s) = t_{oi} \qquad i = 1, 2, \ldots, n \tag{2.5}$$

The problem of calculating the transient temperature distribution in a bearing arrangement thus becomes a problem of solving a system of non-linear differential equations of the first order with certain initial values given. The equations are non-linear since they contain terms of radiation and free convection, which are non-linear with temperature as will be shown later. The simplest and most economical way of solving these equations is to calculate the rate of temperature increase at the time $T = T_k$ from equation 2.4 and then calculate the temperatures at time $T_k + \Delta T$ from

$$t_{k+1} = t_k + \frac{dt_k}{dT} \Delta T = t_k + \frac{q_k}{\rho CpV} \Delta T \tag{2.6}$$

If the time step ΔT used as program input is chosen too large, the temperatures will oscillate, and if it is chosen too small the calculation will be costly. It is therefore desirable to choose the largest possible time step that does not give an oscillating solution. The program optionally calculates such a time step. The step is obtained from the condition, {16}

$$\frac{dt_{i,k+1}}{dt_{i,k}} \geq 0 \qquad i = 1, 2, \ldots, n \tag{2.7}$$

If this derivative were negative, the implication would be that the local temperature at node i has a negative effect on its future value. This would be tantamount to asserting that the hotter a region is now, the colder it will be after an equal time interval. An oscillating solution would result.

Differentiating equation (2.6) for node i, one obtains

$$\frac{dt_{i,k+1}}{dt_{i,k}} = 1 + \frac{\Delta T_i}{\rho_i C_{pi} V_i} \cdot \frac{dq_{i,k}}{dt_{i,k}} \qquad i = 1,2,\ldots n \qquad (2.8)$$

The derivative dq/dt_i is calculated numerically

$$\frac{dq_{i,k}}{dt_{i,k}} = \frac{q_{i,k+1} - q_{i,k}}{\Delta t_i} \qquad (2.9)$$

For each node, the value of ΔT is calculated which gives a value of zero to the right hand side of Eqn. (2.8) and thus Eqn. (2.7) as well.

A value of T rounded off to one significant digit smaller than the ΔT_i given by Eqn. (2.8) is used.

2.1.3. Calculation of Heat Transfer Rate

The transfer of heat within a medium or between two media can occur by conduction, convection, radiation and fluid flow.

All these types of heat transfer occur in a bearing application as the following examples show.

1. Heat is transferred by conduction between inner ring and shaft and between outer ring and housing.
2. Heat is transferred by convection between the surface of the housing and the surrounding air.
3. Heat is transferred by radiation between the shaft and the housing.
4. When the bearing is lubricated and cooled by circulating oil, heat is transferred by fluid flow.

Therefore, in calculating the net flow to a node all the above mentioned modes of heat transfer will be considered.

2.1.3.1 Generated Heat

There may be a heat source at node i giving rise to a heat flow to be added to the heat flowing from the neighboring nodes.

In the case that the heat source is a bearing, it may either be considered to produce a known amount of heat, in which case constant numbers are entered as input to the program, or the shaft-bearing program may be used to calculate the bearing generated heat as a function of bearing temperatures.

2.1.3.2 Conduction

The heat flow $q_{ci,j}$ which is transferred by conduction from node i to node j, is proportional to the difference in temperature $(t_i - t_j)$ and the cross sectional area A and is inversely proportional to the distance ℓ between the two points, thus

$$q_{ci,j} = \frac{\lambda A}{\ell} (t_i - t_j) \qquad (2.10)$$

where λ = the thermal conductivity of the medium.

2.1.3.3 Free Convection

Between a solid medium such as a metallic body and a liquid or gas, heat transfer is by free or forced convection. Heat transfer by free convection is caused by the setting in motion of the liquid or gas as a result of a change in density arising from a temperature differential in the medium. With free convection between a solid medium and air, the heat energy $q_{vi,j}$ transferred between nodes i and j can be calculated from the equation, (2.11)

$$q_{vi,j} = \alpha_v A (t_i - t_j)^d \cdot \mathrm{SIGN}(t_i - t_j) \qquad (2.11)$$

where α_v = the film coefficient of heat transfer by free convection
A = the surface area of contact between the media
d = is an exponent, usually = 1.25, but any value can be specified as input to the program

$$\mathrm{SIGN} = \begin{cases} 1 \text{ if } t_i \geq t_j \\ -1 \text{ if } t_i < t_j \end{cases}$$

The last factor is included to give the expression $q_{vi,j}$ a correct sign.

The value of α_v can be calculated for various cases, see Jacob and Hawkins, (16).

2.1.3.4 Forced Convection

Heat transfer by forced convection takes place when liquid or gas moves around a solid body, for example, when the liquid is forced to flow by means of a pump or when the solid body is moved through the liquid or gas. The heat flow $q_{wi,j}$ transferred by forced convection can be obtained from the following equation.

$$q_{wi,j} = \alpha_w A(t_i - t_j) \tag{2.12}$$

where α_w is the film coefficient of heat transfer during forced convection. This value is dependent on the actual shape, the surface condition of the body, the difference in speed, as well as the properties of the liquid or gas.

In most cases, it is possible to calculate the coefficient of forced convection from a general relationship of the form,

$$N_u = aR_e^{\,b}P_r^{\,c} \tag{2.13}$$

where a, b, and c are constants obtained from handbooks such as (17). R_e and P_r are dimensionless numbers defined by

N_u = Nusselt number = $\alpha_w L/\lambda$
L = characteristic length
λ = conductivity of the fluid
R_e = Reynold's number = $UL\rho/\eta$
U = characteristic speed
ρ = density of the fluid
η = dynamic viscosity of the fluid
P_r = Prandtl number = $\eta C_p/\lambda$
C_p = specific heat

The program can use a value of the coefficient of convection, or let it vary with actual temperatures, the variation being determined by how the viscosity varies. Input can be given in one of the following ways, for each coefficient.

Constant viscosity

1. Values of the parameters of equation (2.13) are given as input and a constant value of α_w is calculated by the program.

Temperature dependent viscosity

2. The coefficient α_w for turbulent flow and heating of petroleum oils is given by

$$\alpha_w = k_9 \cdot \eta(t)^{k_{10}} \tag{2.14}$$

where k_9 and k_{10} are given as input together with viscosity at two different temperatures.

3. Values of the parameters of equation (2.13) are given as input together with viscosity at two different temperatures.

2.1.3.5 Radiation

If two flat parallel, similar surfaces are placed close together and have the same surface area A, the heat energy transferred by radiaiton between nodes i and j representing those bodies, will be,

$$q_{Ri,j} = \varepsilon\sigma A \; (t_i+273)^4 - (t_j+273)^4 \tag{2.15}$$

where ε is the surface emissivity. The value of the coefficient ε is an input variable and varies between 1 for a completely black surface and 0 for an absolutely clean surface. In addition σ is Stefan-Boltzmann's radiation constant which has the value 5.76 x 10^{-8} watts/m^2-$(^oK)^4$ and t_i and t_j are the temperatures (°C) at points i and j.

Heat transfer by radiation under other conditions can also be calculated, (16). The following equation, for instance applies between two concentric cylindrical surfaces.

$$q_{Ri,j} = \frac{\varepsilon\sigma A_i\left[(t_i+273)^4 - (t_j+273)^4\right]}{1 + (1-\varepsilon)\ (A_i/A_e)} \qquad (2.16)$$

where A_i is the area of the inner cylindrical surface
A_e is the area of the outer cylindrical surface

2.1.3.6 Fluid Flow

Between nodes established in fluids, heat is transferred by transport of the fluid itself and the heat it contains.

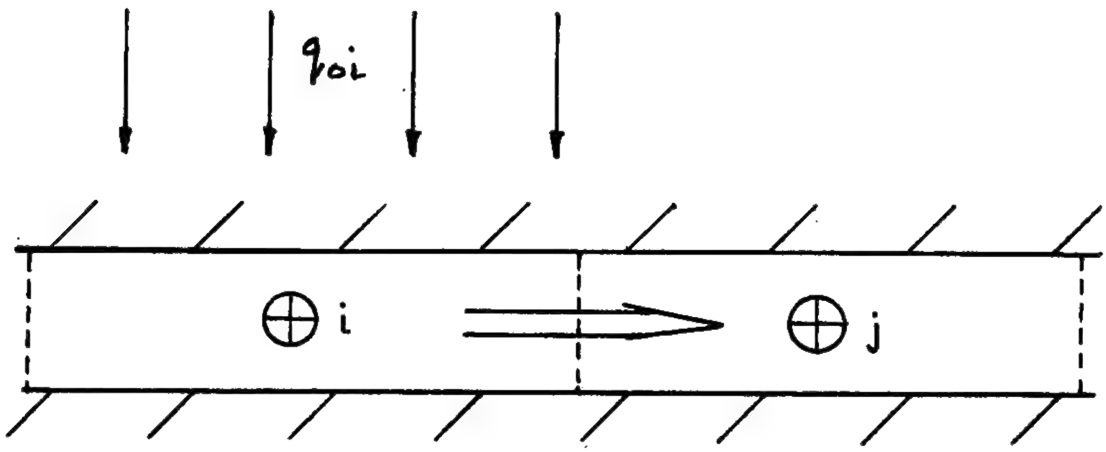


Figure 2.1 Convective Heat Transfer

Figure 2.1 shows nodes i and j at the midpoints of consecutive segments established in a stream of flowing fluid.

The heat flow $q_{ui,j}$ through the boundary between nodes i and j can be calculated as the sum of the heat flow q_{fi} through the middle of the element i, and half the heat flow q_{oi} transferred to node i by other means, such as convection.

The heat carried by mass flow is,

$$q_{fi} = \rho_i\ C_{p_i}\ V_i\ t_i = K_i t_i \qquad (2.17)$$

where V_i = the volume flow rate through node i

The heat input to node i is the sum of the heat generated at node i (if any) and the sum over all other nodes of the heat transferred to node i by conduction, radiation, free and forced convection.

$$q_{oi} = q_{G,i} + \sum_j (q_{ci,j} + q_{vi,j} + q_{wi,j} + q_{Ri,j}) \tag{2.18}$$

The heat flow between the nodes of Fig. 2.2 is then

$$q_{ui,j} = q_{fi} + q_{oi}/2 \tag{2.19}$$

If the flow is dividing between node i and j, Figure 2.2 then the heat flow is calculated from

$$q_{ui,j} = K_{ij}\ (q_{fi} + q_{oi}/2) \tag{2.20}$$

where K_{ij} = the proportion of the flow at i going to node j, $0 < K_{ij} \leq 1$. K_{ij} is specified at input.

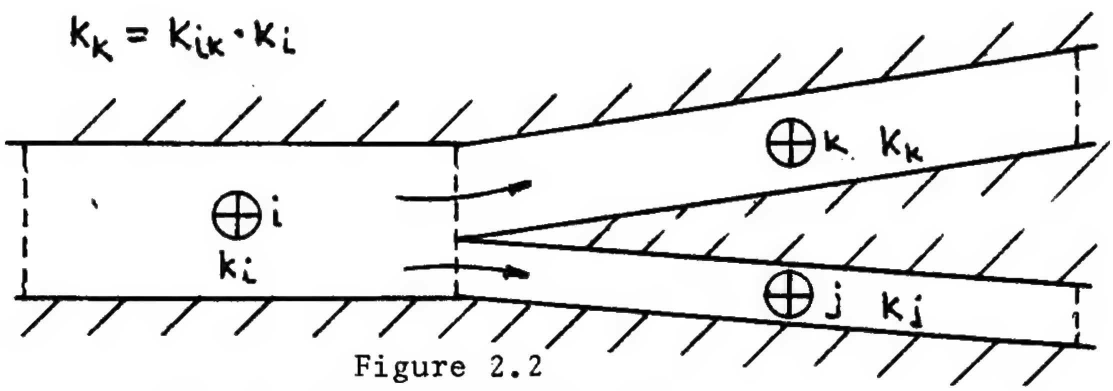


Figure 2.2
DIVIDED FLUID FLOW FROM NODE i

2.1.3.7 Total Heat Transferred

The net heat flow rate to node i can be expressed as,

$$q_i = q_{G,i} + \sum_j (q_{ci,j} + q_{ui,j} + q_{vi,j} + q_{wi,j}) \tag{2.21}$$

The summation should include all nodes j, both with unknown temperatures as well as boundary nodes, at which the temperature is known so long as they have a direct heat exchange with note i.

This expression is a non-linear function of temperatures because of the terms q_W and q_R. Therefore the equations to be solved for a steady state solution are non-linear.

2.1.4 Conduction Through a Bearing

As described in Section 2.1.3.2 the conduction between two nodes is governed by the thermal conductivity parameter λ of the medium through which conduction takes place. The value of λ is specified at input.

An exception is when one of the nodes represents a bearing ring and the other a set of rolling elements. In this case the conduction is separately calculated using the principles described below. Note that separate calculations are performed for the rolling element raceway contacts and the rolling element-flange contacts. The methods for both calculations are identical and are performed within the program.

2.1.4.1. Thermal Resistance

It is assumed that the rolling speeds of the rolling elements are so high that the bulk temperature of the rolling elements are the same at both the inner and outer races, except in a volume close to the surface. The resistance to heat flow can then be calculated as the sum of the resistance across the surface and the resistance of the material close to the surface.

The resistance Ω is defined implicitly by

$$\Delta t = \Omega \cdot q \tag{2.22}$$

where

Δt is temperature difference
q is heat flow

The resistance due to conduction through the EHD film is calculated as

$$\Omega_1 = (h/\lambda) \cdot A \tag{2.23}$$

where h is taken to be the calculated plateau film thickness
A is the Hertzian contact area at the specific rolling element-ring contact under consideration.
λ is the conductivity of the oil.

The geometry is shown in Figure 2.3(a). Asperity conduction is not considered.

So far, a constant temperature difference between the surfaces has been assumed. But during the time period of contact, the difference will decrease because of the finite thermal diffusivity of the material near the surface, Fig. 2.3 (b).

Two points at a distance from the surface this phenomenon will have the same effect as an additional resistance Ω_2 acting in series with Ω_1.

This resistance was estimated in {18} as,

$$\Omega_2 = \frac{1}{\lambda \ell_{re,i}} \left(\frac{\pi\psi}{2b_i V}\right)^{\frac{1}{2}} \tag{2.24}$$

where ℓ_{re} = contact length, or inthe case of an elliptical contact area, 0.8 times the major axis

λ = heat conductivity

ψ = thermal diffusivity = $\lambda/(\rho.C_p)$

ρ = density

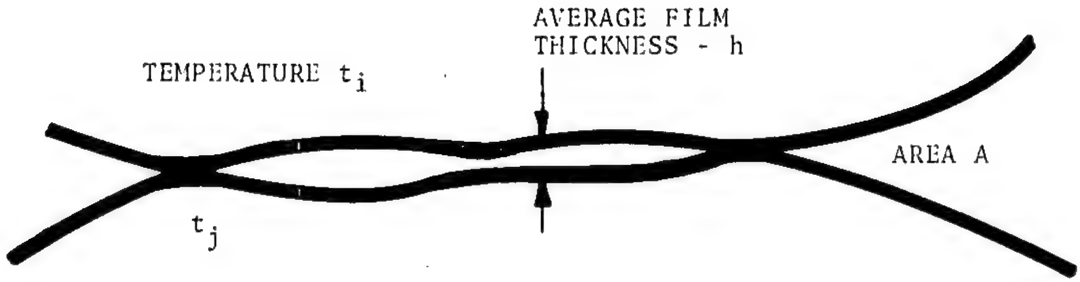
C_p = specific heat

b = half the contact width

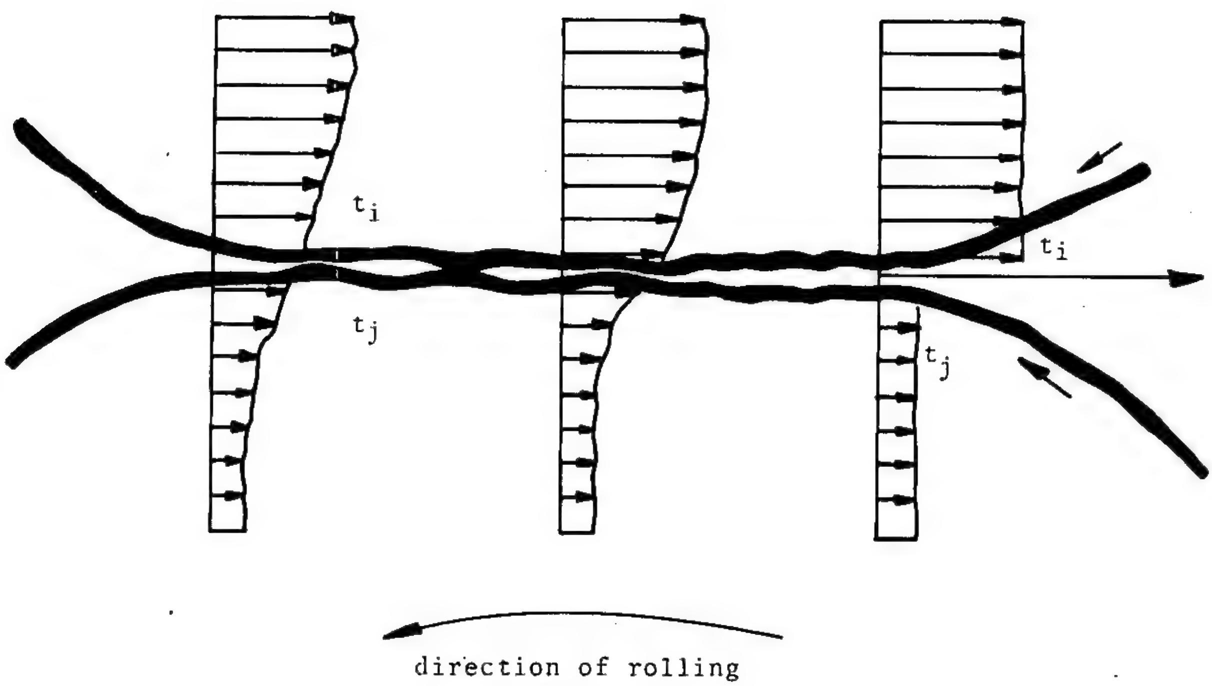
V = rolling speed

The resultant resistance is

$$\Omega_{res} \quad \Omega_1 + \Omega_2$$



(a) Schematic Concentrated Contact



(b) Temperature Distribution at Rolling, Concentrated Contact Surfaces

FIGURE 2.3

CONTACT GEOMETRY AND TEMPERATURES

There is one such resistance at each rolling element. They all act in parallel. The resultant resistance Ωres, is thus obtained from

$$\frac{1}{\Omega_{res}} = \sum_{i=1}^{n} \frac{1}{\Omega_{res,i}} \tag{2.26}$$

2.2 Bearing Dimensional Change Analysis

The program calculates the changes in bearing diametral clearances according to the analysis described in {5}, and expressed in generalized equation form as,

$$\Delta DCL = f\{(Fits)_m,\ t_i,\ {}_m,\ (Q_r)_m\}, \tag{2.27}$$

m = 1,2 for inner and outer rings respectively

i = 1,2,3,4,5 for shaft, inner ring, outer ring, housing and rolling element respectively

where: ΔDCL is the change in bearing diametral cearance
Fits are the cold mounted shaft and housing fits.
t_i are the component temperatures
Ω_m refers to the ring rotational speeds
Q_r^m refers to the radial component of the minimum rolling element-race normal force

A bearing clearance change criterion is satisfied when the change in bearing diametral clearance remains within a narrow, user specified range, for two successive iterations as follows:

$$\frac{|(\Delta DCL)_N - (\Delta DCL)_{N-1}|}{D} \leq \text{EPSFIT for all bearings} \quad (2.28)$$

where: N denotes the most recent iteration and
N-1 denotes the previous iteration,
D denotes the ball or roller diameter and
EPSFIT is a user specified value, = .0001D

It should be noted that although ring rotational speeds, and initial, i.e. cold, shaft and housing fits are considered in the clearance change analysis, these two factors are fixed at input and remain constant through the entire solution. Although component temperatures may change as a consequence of the thermal solution, temperatures remain constant through a complete set of clearance change iterations. As a result, only the change in bearing load distribution affects the change in bearing clearance within a set of clearance change iterations.

2.3 Bearing Inner Ring Equilibrium

The bearing inner ring equilibrium solution is obtained by solving the system:

$$(\vec{FM}_b)_i - (\vec{FM}_s)_i = 0 \text{ for all bearings, } i \quad (2.29)$$

where: $\vec{FM}_b$ denotes a vector of bearing loads and moments resulting from rolling element/race forces and moments.

$$\vec{FM}_{bi} = \begin{bmatrix} F_{bxi} \\ F_{byi} \\ F_{bzi} \\ \hline M_{byi} \\ M_{bzi} \end{bmatrix} \begin{matrix} \text{Forces} \\ \\ \text{Moments} \end{matrix} \tag{2.30}$$

If the bearing solution considers friction, $\vec{FM}_b$ is comprised of the ball race friction forces as well as the normal forces.

If the bearing solution is, at the user's option, frictionless, $\vec{FM}_b$ is comprised only of rolling element/race normal contact forces.

$\vec{FM}_{si}$ denotes a similar vector of loads, exerted on the inner ring by the shaft.

$$\vec{FM}_{si} = \begin{bmatrix} F_{sxi} \\ F_{syi} \\ F_{szi} \\ \hline M_{syi} \\ M_{szi} \end{bmatrix} \begin{matrix} \text{Forces} \\ \\ \text{Moments} \end{matrix} \tag{2.31}$$

The variables in this system of equations are the bearing inner ring deflections $\vec{\Delta}_b$ and the shaft displacements $\vec{\Delta}$ at all bearing locations. The bearing loads may be expressed as a function of the inner ring deflections.

$$\vec{FM}_b' = \vec{FM}_b \; (\vec{\Delta}_b) \tag{2.32}$$

The deflection $\vec{\Delta}_b$ of a bearing is described by two radial deflections δ_y and δ_z, two angular deflections θ_y and θ_z and axial deflection δ_x. The axial deflection is assumed to be the same for all bearings.

The solution scheme is ended when

$$\frac{\delta(\vec{\Delta})_{ij}}{(\vec{\Delta})_{ij}} \leq \begin{array}{l} \text{EPS1 (frictionless)} \\ \text{EPS2 (friction)} \end{array} \qquad (2.37)$$

i = 1,... (Number of bearings)

j = 1,5 for the 3 linear and two angular deflections at each bearing

If for some i or j, $(\vec{\Delta})_{ij} = 0$, Eq (2.38) is used in place of (2.37).

$$\frac{\delta(\vec{\Delta})_{ij}}{(0.001 \times \text{NBRG})} - \begin{array}{l} \text{EPS1(frictionless)} \\ \text{ESP2 (friction)} \end{array} \qquad (2.38)$$

NBRG denotes the number of bearings in the system.

EPS1 or EPS2 is used depending on whether the bearing solutions are fricitonless or include friction, respectively. If the bearing deflections are extremely small, computer-generated numerical inaccuracies may prevent convergence according to the above criteria although a perfectly good solution has been obtained. To overcome this problem, the iteration is terminated if all angular deflections are less than 2×10^{-6} radians and all linear deflections are less than 5×10^{-8} inches. Any one of the above criteria imply that inner ring equilibrium is satisfied.

2.4 Bearing Quasi-Dynamic Solution

The bearing quasi-dynamic solution is obtained through a two step process:

1) Elastic Solution - considering rolling element centrifugal force, plus the gyroscopic moment for a tapered roller.

2) Elastic and Quasi-dynamic Solution*

*Quasi-dynamic equilibrium is used to connote that the true dynamic equilibrium terms containing first derivatives of the ball rotational speed vectors and the second derivatives of rolling element position vectors with respect to time are replaced by numerical expressions which are position rather than time dependent.

The equations which define rolling element quasi-dynamic force equilibrim take the form

$$\sum_m \left[\int_{-a_m}^{a_m} (\vec{Q}_m + \vec{f}_m) ds + \vec{F}_m \right] + \vec{F} = 0 \qquad (2.39)$$

m = 1-3 refers to the outer inner and cage rolling element contacts respectively
m = 1-4 for tapered rollers with one roller end-flange contact.
m = 1-7 for cylindrical rollers where four roller and flange contacts are possible.

where: $\vec{Q}_m$ is the vector normal load per unit length of the contact. See Ref. (1).

f_m is the vector of friction force per unit length of the contact. See Ref. (9).

F is the vector of inertia and drag forces. See Ref. (1)

s is a coordinate along the contact perpendicular to the direction of rolling (usually the major axis)

a is half the contact length. See Ref. (1).

F_m is the vector sum of the hydrodynamic forces acting on the rolling element at the m-th contact. For ball-raceway contact see Ref. (9). For the roller-raceway contact, see Ref. (16).

Rolling element moment equilibrium is defined by:

$$\sum_m \left[\int_{-a_m}^{a_m} \vec{r}_m \cdot (\vec{Q}_m + \vec{f}_m) ds \right] + \vec{r}_m \cdot \vec{F}_m + \vec{M}_I = 0 \qquad (2.40)$$

$\vec{Q}_m, \vec{f}_m, \vec{F}_m, \vec{a}_m$, and s are defined above, $\vec{M}_I$ is a vector of inertia moments. For the definition of M_I refer to Ref. (1).

$\vec{r}_m$ is a vector from the rolling element center to the point of contact.

In the frictionless elastic solution $\vec{F}_m$ and $\vec{f}_m = 0$. Additionally, the only rolling element inertia term considered in the frictionless solution is centrifugal force, plus the gyroscopic moment for tapered rollers. As a consequence only the axial and radial force equilibrium equations are solved for each ball. For each roller the radial and axial force equilibrium and the tilting moment about the z axis of Fig. 2.4 is solved A dummy equation for axial force equilibrium is included in the

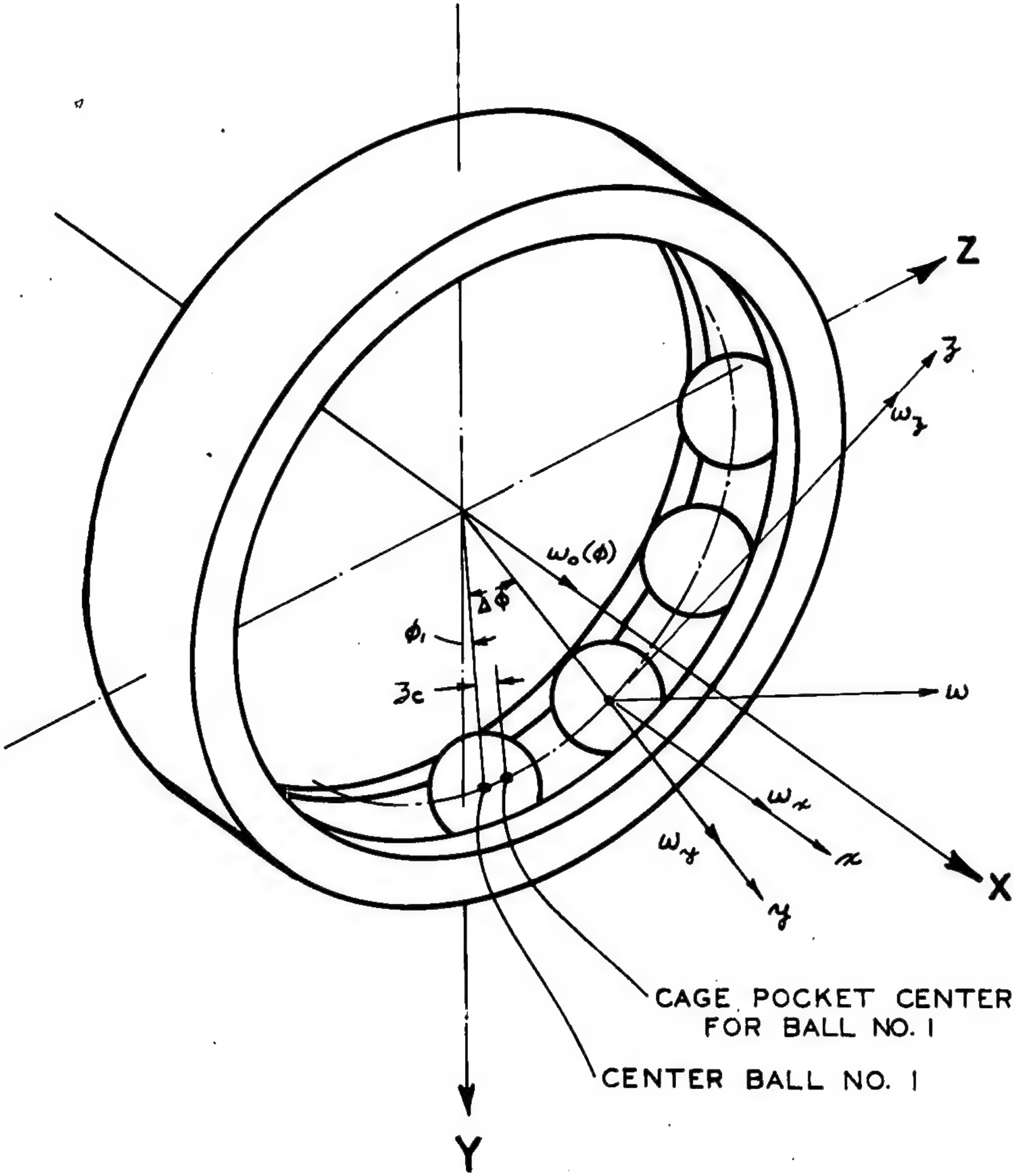


FIGURE 2.4

BEARING INERTIAL (XYZ) AND ROLLING ELEMENT (xyz), COORDINATE SYSTEMS

solution matrix which keeps the roller centered with respect to the outer race if the cylindrical roller bearing has no flanges.

The friction solution determines ball quasi-dynamic equilibrium for six degrees of freedom. A roller is permitted four degrees of freedom. The rolling-element variables in this solution are xl, yl, Wx, Wy, Wz, and Wo.

where x_1 is the rolling element axial position relative to the outer race.
y_1 is the rolling element radial position relative to the outer race,
Wx, Wy, Wz are orthogonal rolling element rotational speeds relative to the cage speed, about the x,y, and z z axes and W_o is the rolling element orbital speed. For the roller, W_z is a dummy variable.

The variables x_1 and y_1 are the ball variables in the frictionless solution. The variables in the roller frictionless solution are x_1, y_1 and $\Theta z = \tan - 1\ (W_y/W_x)$.

The details of the cage analysis are contained in SKF Report No. AL76P003. Either one or three cage equilibrium equations are considered depending upon cage type and the degree of rolling element orbital speed variation.

The cage equations and cage-rolling element interactions are not considered when the friction forces are omitted from the rolling element equilibrium equations.

The number of degrees of freedom given to the cage is one, if the cage will tend to rotate concentrically with respect to the ring on which it is riding. This condition is determined as a function of the rolling element orbital speed variation and prevails with most roller bearings and with ball bearings subjected only to axial loading. In both cases, orbital speed variation is often inconsequential. Also, a single degree of freedom is allowed when the cage is rolling element riding. The single degree of freedom corresponds to a small angular rotation about the bearing axis, measured with respect to rolling element 1. The angular displacement is converted to a linear dimension by a multiplication by the bearing pitch diameter and is noted in Fig. 2.4 as ξc. When a single degree of freedom is permitted, the sum of moments acting on the cage about the bearing x axis is required to be zero. This moment equation considers the cage-rolling element normal and friction forces as well as the torque generated at the cage-ring surface.

If there is significant rolling element orbital speed variation, the cage is permitted to move to an eccentric position with respect to the land on which it is piloted. Two additional degrees of freedom are required to describe the eccentric position. These additional degrees of freedom are the cage center of mass radial displacement, e, and the angular displacement of the center of mass, with respect to the bearing Y axis, Θ_c^1, See Figures (2-5 and 2-6). These radial

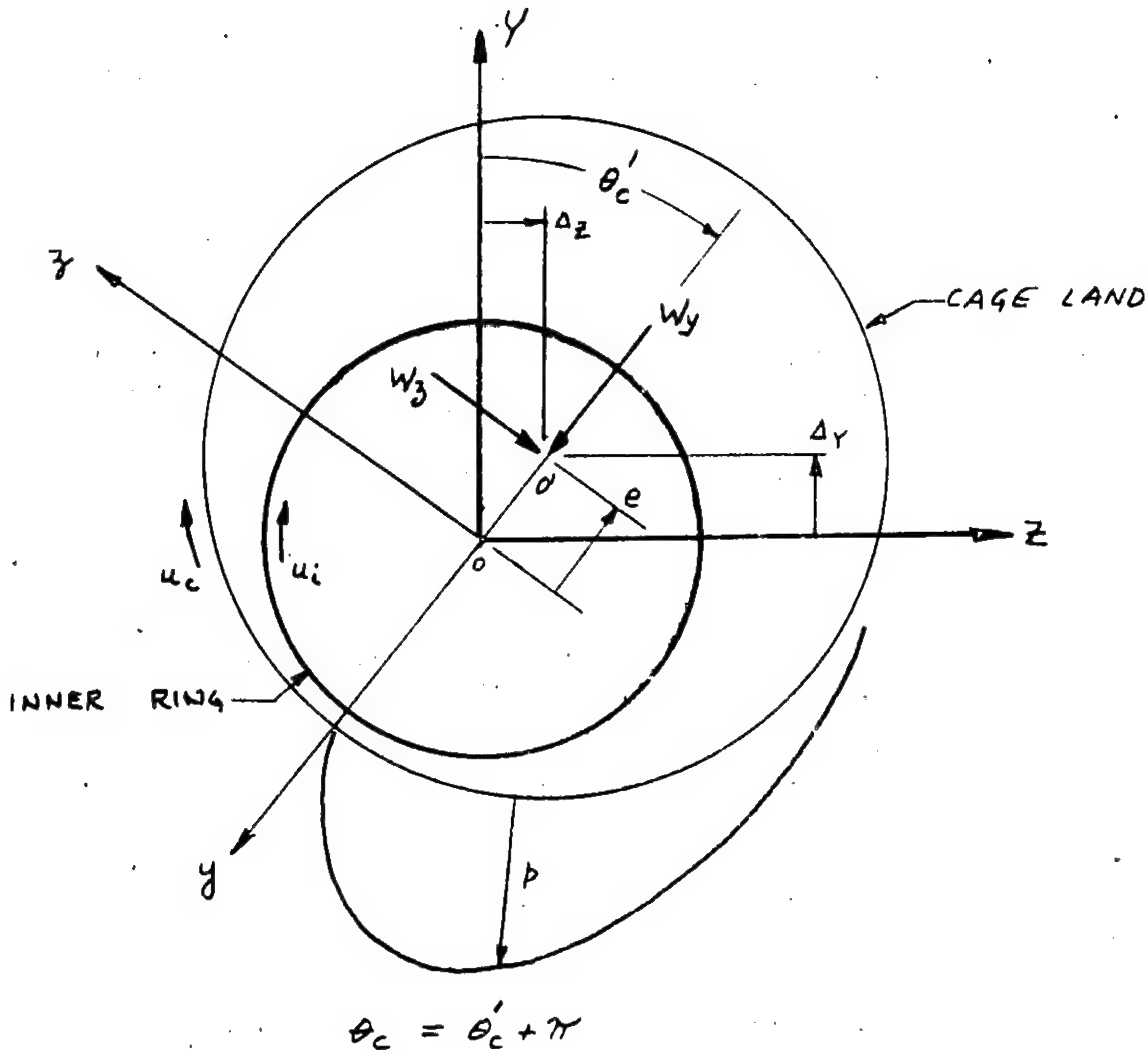
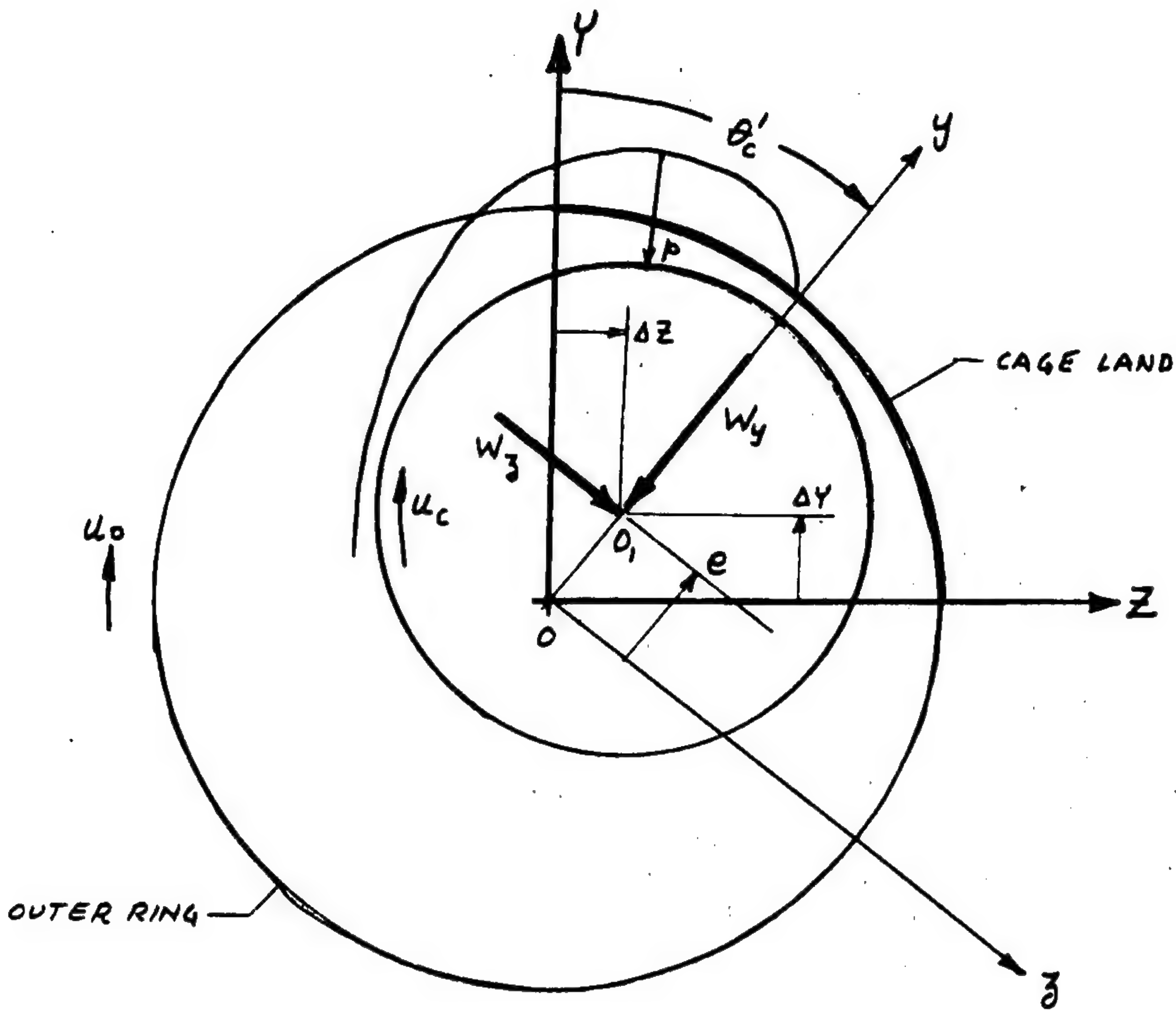


Figure 2- 5 : Inner Ring-Cage Land Contact Geometry



$\theta_c = \theta_c'$

Figure 2-6 : Outer Ring-Cage Land Contact Geometry

friction forces as well as the pressure buildup between the cage and its piloting surface are considered in the equilibrium equations. The effect of the cage mass is neglected.

The ball bearing friction solution is thus obtained by soving 6Z+(1 or 3) equations where Z is the number of rolling elements. The ball bearing frictionless solution is obtained by solving 1, (Z/2) (Z/2+1) or Z sets of 2 equations, depending upon the number of rolling elements in the bearing and the degree of load symmetry which prevails. The various symmetry conditions are explained below.

The roller bearing friction solution contains 4Z+(1 or 3) equations and the frictionless solution contains Z/2, Z/2+1 or Z sets of three equations again depending upon the number of rolling elements and whether or not load symmetry exists.

The various load symmetry conditions are as follows. Axial symmetry is utilized if the load is axial only, then only one set of two or three equations is solved for the frictionless case and six rolling elements and one cage equilibrium equation is solved when friction is included. All rolling elements are assumed to behave identically.

Radial load symmetry is utilized if the non-axial shaft loading is comprised of only radial components parallel to the Y axis and moment components parallel to the Z axis and the position of the first rolling element is utilized. When this symmetry exists, only half the rolling elements need be considered if the number of rolling elements is even and one half plus one need be considered if the number is odd. Because of inertia terms, radial load symmetry can only be utilized in the frictionless calculations.

If load symmetry is not present, then Z sets of two (ball bearing) or Z sets of three (roller bearing) equations must be solved to obtain the frictionless solution.,

As with the steady state temperature mapping scheme, the Newton-Raphson scheme in subprogram SOLV13 is used to solve the sets of equations for each bearing. The iteration scheme terminates when either:

$$\left| \frac{X_{in}}{X_{in-1}} \right|_{i=1\ldots n} < \begin{cases} \text{EPS1} & \text{frictionless} \\ \text{EPS2} & \text{friction} \end{cases} \qquad (2.41)$$

or

$$\left[\frac{\sum_{i=1}^{n} EQ_i^2}{n} \right]^{\frac{1}{2}} < 100^* \begin{cases} \text{EPS1} & \text{frictionless} \\ \text{EPS2} & \text{friction} \end{cases} \qquad (2.42)$$

Experience has shown that the second criterion is usually responsible for terminating the solution. However, when rolling element loads are extremely large, on the order of 10^5 Newtons, it becomes difficult to reduce the equation residues to less than 10 Newtons. In those instances, the first criteria usually terminates the iteration scheme.

3. PROGRAM INPUT

3.1 Types of Input Data

A complete set of input data is comprised of four distinct categories of data. Within these categories, cards which convey specific kinds of information are referred to as card types. Depending on the complexity of the problem, the input data set may contain none, one or several cards of a given type. The categories are listed below.

I. Title Cards
   A title card plus a second card which provides the program control information for the shaft-bearing solution.

II. Bearing Data Cards
   A set of up to sixteen (16) card types. Each set describes one bearing in the assembly. All bearings must be so described. The card sets must be input sequentially in order of increasing distance from a selected end of the shaft.

III. Thermal Data Cards
   A set of up to nine (9) card types to describe the thermal model of the assembly.

IV. Shaft Data Cards
   A set of three (3) card types to describe the shaft geometry, bearing locations on the shaft and shaft loading.

If the program is being used to predict the performance of a bearing assembly, cards from all four sets must be included in the runstream. If the program is being used to thermally model a mechanical system wherein no bearing heat generation rates are required, and therefore no bearing calculations need be performed, the cards from sets II and IV are omitted.

The review of required input information which follows is broken into the four sets of data categories given above, with special emphasis on program control data.

The input data instructions are given in Appendix C, and are for the most part, self explanatory. They are laid out in the format of an eighty column data card. A description of the variables is given in the input instruction forms.

The units used for input data are as follows:

Linear Dimensions - (mm)
Angles - (degrees)
Surface Roughness (microns)
Bearing Angular Mounting Errors - (radians)
Rotational Speeds - (RPM)
Force - (Newtons) (N)
Moments - (N-mm)
Pressure, Elastic Modulus - (N/mm^2)
Density - (gm/cm^3)
Kinematic Viscosity - (cs)
Temperature - (degrees centigrade) (^oC)
Coefficient of Thermal Expansion - ($^oC^{-1}$)
Thermal Conductivity - (Watts/m/^oC)

3.2 Data Set L - Title Cards

3.2.1 Title Card 1

This card should contain the computer run title and any information which might prove useful for future identification. The full eighty (80) columns are available for this purpose. The title will appear at the top of each page of Program output.

3.2.2 Title Card 2

This card provides the control information for the shaft bearing solution.

Item 1: Shaft Speed in rpm, GOV (1). All bearings have the same shaft, inner ring, and speed.

Item 2: Number of Bearings on the shaft (NBRG), a minimum of zero is permitted if no bearing solution is being sought. A maximum of five is permitted. Note that a bearing is defined as a single row of rolling elements. Thus a double row bearing is treated as two separate, single row bearings.

Item 3: Print Flag (NPRINT), NPRINT equal to zero is normal and will result in no intermediate or debug output. With a value of one, a low level intermediate print is obtained at the end of each shaft bearing iteration. The values of the variables, the inner ring displacements (DEL), and the equation residues are printed.

At the end of each bearing iteration, wherein the rolling element and cage equilibrium equations are solved, an error parameter is printed which has the value:

$$\text{Error Parameter} = \Delta X_N / X_{N-1}$$

ΔX_N is the change in the variable X specified at iteration N.

X_{N-1} is the value of the variable specified at the previous iteration.

The Error Parameter is calculated for each of the bearing variables, but only the largest one is printed.

Additionally, at the end of each Clearance Change iteration, the clearance change error parameter is printed. This error is defined:

$$\text{Error Parameter} = \frac{DCL_N - DCL_{N-1}}{\text{Rolling Element Diameter}}$$

where DCL_N and DCL_{N-1} denote the clearance changes calculated at the current and previous iterations respectively.

If NPRINT is set at 2 all of the above information is printed. Additionally the variable values and residue values are printed for each iteration of the rolling element and cage equilibrium solution.

Item 4: ITFIT controls the number of iterations allowed to satisfy the bearing clearance change iteration scheme. If ITFIT is set to zero (0), or left blank, the clearance change portion of the program is not executed. If a positive integer is input, the clearance change scheme is utilized with a maximum iteration limit of five (5). If a negative integer is input, the scheme is used with a maximum iteration limit equal to the absolute value of the negative integer.

Item 5: ITMAIN limits the number of iterations attempted during the solution of the shaft and bearing inner ring equilibrium problems for establishing the equilibrium of bearing reactions and applied shaft loads. If ITMAIN is left blank, set to zero, or to a positive integer, then (15) iterations are permitted. If ITMAIN is set to a negative integer the number of iterations is limited to the absolute value of that integer.

Item 6: GOV(2) or EPSFIT is the convergence criterion for the diametral clearance change portion of the analysis. As mentioned under item 3 above, this error parameter is defined by Eq. 2.28.

The iteration scheme is terminated when the error parameter is less than the input value of EPSFIT. If EPSFIT is left blank or is set to zero (0), the program default value of 0.0001 is used.

Items 7 & 8: Main loop accuracy for frictionless elastic (EPS1) and friction solution (EPS2). These accuracy values control the accuracy of the shaft bearing deflection solution as well as the quasi-dynamic solution of the component dynamics. If EPS1 and EPS2 are left blank or set to zero (0), default values of 0.001 and 0.0001 respectively are used.

Item 9: IMT, if set to 1, the material properties for both bearing rings and the rolling elements are to be input on card types B 11 through B 19. If IMT is zero or blank, the rings and rolling elements are assumed to be 52100 bearing steel. Card types B 11 through B 14 are required only if the change in bearing diametral clearance is to be calculated.

Item 10: NPASS controls the level of the bearing solution:

0 Elastic Contact Forces are calculated. No lubrication or friction effects are considered.

1 Elastic Contact Forces are calculated. Lubrication and friction effects are considered using raceway control (ball bearing) or epicyclic (roller bearing) assumptions to estimate rolling element and cage speeds.

2 Inner Equilibrium is satisfied considering only the Elastic Contact Forces. Using the inner ring positions thus obtained, rolling element and cage equilibrium are determined considering friction.

3 Complete Solution. The inner ring, rolling element and cage equilibrium is determined considering all elastic and friction forces.

3.3 Data Set II - Bearing Data

Most of the input instructions are self-explanatory. Where certain items are deemed to require more explanation than given in the input data format instructions they are treated on an individual basis by card type and item number.

Most of the bearing input data is read into a two dimensional array named "BD", which has the dimensions (1830, 5). For each of the five bearings permitted on a shaft, a total of 1830 pieces of data may be stored. Denoting BD(I,J), I represents a specific piece of bearing data, J represents the bearing number. The bearing input data of Data Set II occupies the first 106 locations of the 1830 allotted. On the input data format sheets the designation BD(I) where I=1...106, denotes the location within the BD array where each piece of input data is stored.

3.3.1 Card Type 1 - Bearing Type and Material Designations

Item 1: Bearing type, columns 1-10 must be specified, left justified, i.e., "B", "C" or "T" in column 1. This format must be followed since the Program recognition of bearing type, (ball, cylindrical or tapered roller bearing), is derived from reading the "B", "C", or "T" in the first column of this card.

Items 2 & 3: Columns 11-30 and 31-50, "Steel designations", inner and outer rings respectively. The alphameric-literal description of the steel types such as "M-50" or "AISI 52100" is input.

Items 4 & 5: Columns 51-60 and 61-70, the numbers input for items 4 and 5 are used to account for improved materials and multiply the raceway fatigue lives as determined by Lundberg-Palmgren methods. Typical life factor values for modern steels are in the neighborhood of 2.0 to 3.0. If the ASME Publication Life Adjustment Factors for Ball and Roller Bearings, is referenced by the user, the Material Factor D and the Material Process Factor E should be used as multipliers as inputs for items 4 and 5. Additionally if the user is accustomed to using a lubricant life multiplier he must also multiply the material factor by the lubricant life multiplier. The program considers EHD film thickness and RMS surface roughness but generates a life multiplier having a maximum value of 1 and a miminum of 0.479, i.e. Lube-Life Factor Programmed only serves to reduce predicted Fatigue Life.

Item 6: Columns 71-78, "Orientation angle of the first rolling element". ($\phi 1$) (degrees). Refer to Fig. (2.4). The quasi-dynamic rolling element bearing problem has an infinite number of solutions which fall within a narrow envelope having a periodic shape. The solution obtained is a function of the rolling element positions relative to the bearing system coordinate axes. $\phi_1 = 0$, places a rolling element on the Y axis and is the choice customarily made. ϕ_1 can be desig-

nated as any value $0<\Phi<\frac{360}{Z}$ where Z is the number of rolling elements. For each different value assigned to $\Phi 1$ a different, although similar, bearing solution will be obtained. To take advantage of bearing symmetry and the computer time savings which result Φ_1 must be specified as zero or left blank.

Item: 7 Column 80, a signal, termed the crown drop flag, which specifies for a cylindrical or tapered roller bearing, whether the roller-race crown drops will be calculated, or read directly. If item 7 is blank or zero, the crown drops are calculated based on the roller-race crown radius, and effective flat length input information. If the crown drop flag is other than zero or blank the non-uniform separation of the roller and raceway must be specified at the center of each slice into which the roller-raceway effective contact length is divided. The slice widths are identical. The number of slices is input as item 7 card type B4. The non-uniform roller-raceway separation is input on card types B5 and B6.

3.3.2 Card Type B2 - Bearing Geometry and Outer Ring Speed

3.3.2.1. Ball Bearing Geometry

Items 1, 2 and 6 are self-explanatory. Item 5 pertains only to a tapered roller bearing, as discussed later. Items 3 and 4 require explanation however.

Through the proper specification of the diametral clearance and contact angle, the program can properly handle deep groove, split inner, and angular contact ball bearings.

The deep groove ball bearing requires the specification of zero contact angle and either the operating diametral clearance Pd or the off the shelf diametral clearance, if the dimensional change analysis is utilized.

The angular contact bearing is fully described through specification of the contact angle measured under axial load. However, this method of input does not accurately define the system if there is more than one angular contact supporting the shaft and at least one of those bearings has its grooves offset in the direction opposite to the other bearings and if the shaft is capable of axial and/or radial play. In other words, if what are known as angular contact ball bearings are mounted such that some diametral shaft play is permitted, an auxilliary angle as well as the diametral play must be specified at input. The angle input is not the

manufacturer's designated contact angle, α, but an auxilliary angle, α_o, the calculation for which shall be demonstrated.

Refer to Figure 3.1. The manufacturer's contact angle is calculated as follows:

$$\alpha = \cos^{-1}\left[\frac{2A - Pd}{2A}\right] \tag{3.1}$$

$$A = r_o + r_i - D \tag{3.2}$$

where: r_o and r_i are the outer and inner raceway groove radii respectively

D is the ball diameter

Under a gauge axial load, α is obtained at both inner and outer raceways for each ball. Under this condition, the outer and inner raceways are axially offset an amount S_α.

$$S_\alpha = A \sin\alpha \tag{3.3}$$

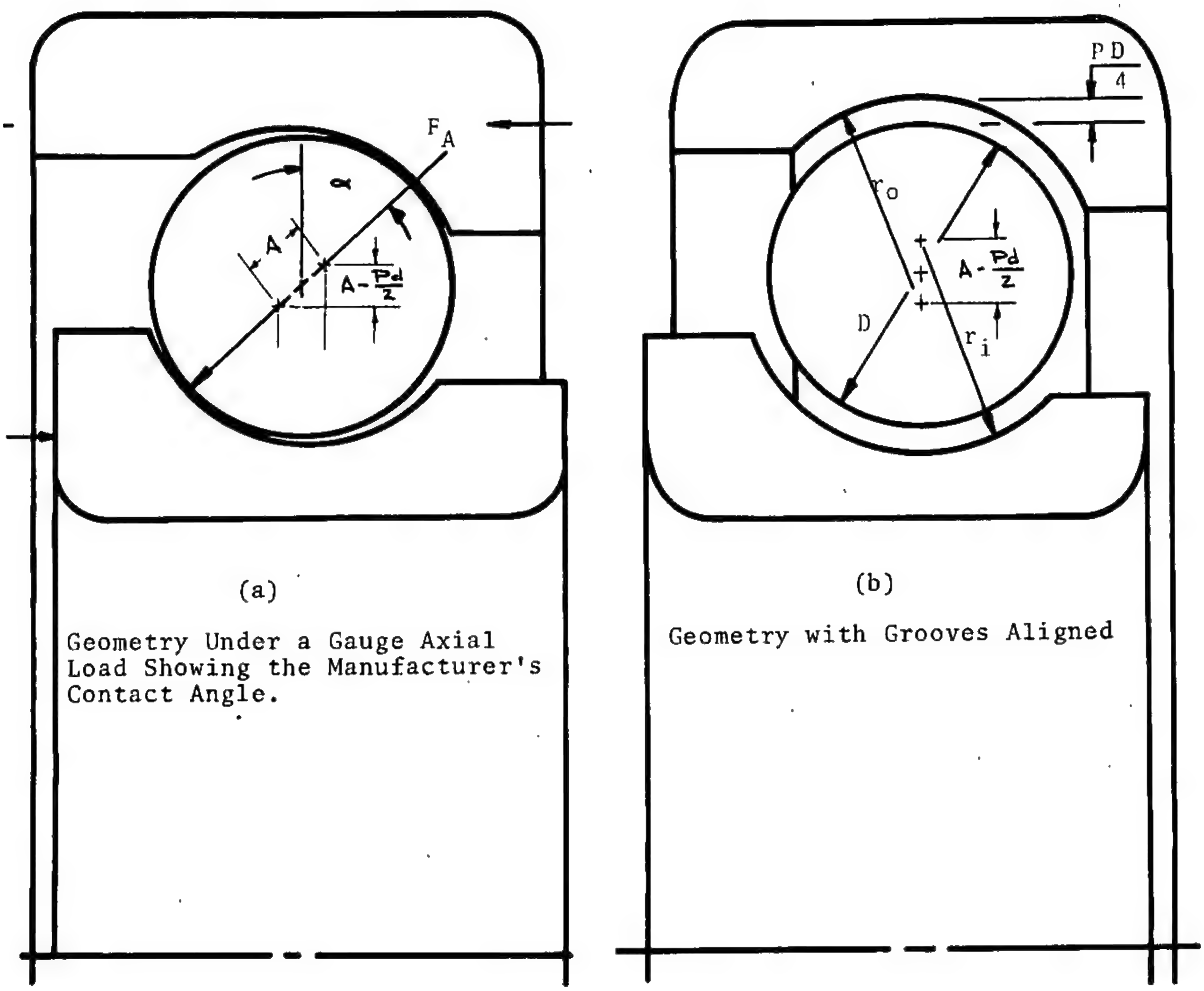
When angular contact ball bearings are mounted with some diametral play, the grooves are offset an amount $S\alpha_o$ such that $S\alpha_o < S\alpha$. The diametral play which obtains at this condition is Sd. This diametral play is usually known by the engineer or designer and is usually required to allow some forgiveness when thermal gradients are encountered. Assuming that the user has the values for α, r_o, r_i, D and $S\alpha_o$ then:

$$\alpha_o = \tan^{-1}\left[\frac{S\alpha_o}{A - \frac{Pd}{2}}\right] \tag{3.4}$$

where: Pd and A may be calculated from Eqs. (3.1) and (3.2).

If $S\alpha_o$ is unknown, the following equation may be solved for α_o.

FIG. 3.1 ANGULAR CONTACT BALL BEARING GEOMETRY



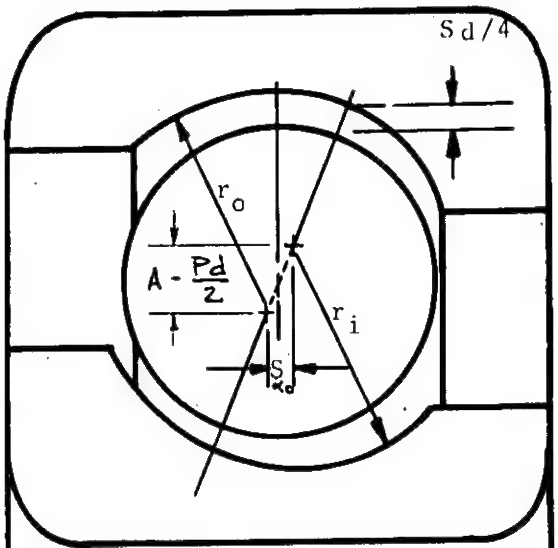
(a)

Geometry Under a Gauge Axial Load Showing the Manufacturer's Contact Angle.

(b)

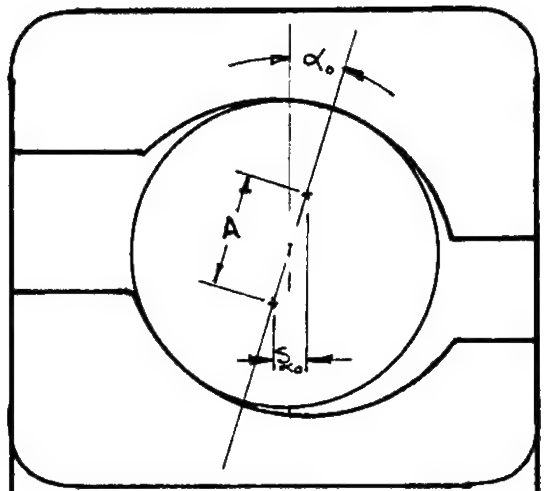
Geometry with Grooves Aligned

FIG. 3.1 ANGULAR CONTACT BALL BEARING GEOMETRY



(c)

Geometry Showing the as Mounted Groove Alignment and the Input Diametral Play (Sd).



(d)

Geometry Showing Groove Alignment with Rings Subjected to a Gage Radial Load, i.e., with the Radial Play Removed. Also shown, the Auxilliary Input Contact Angle α_o

$$\alpha_o = \cos^{-1}\left[\frac{2A - Pd + Sd}{2A}\right] \quad (3.5)$$

In order that the Program properly handle split inner ring ball bearings an auxilliary angle and diametral play must be input. Referring to Figure 3.2, the auxilliary angle α_o and diametral play Sd must be determined and input. Typically the values of D, r_o, r_i, α_s and Sd' are known and Pd may be calculated as follows:

$$Pd = Sd' + (2r_i - D)(1 - \cos\alpha_s) \quad (3.6)$$

The unloaded half of the inner ring must be removed from consideration and the ball moved such that its center lies on the line connecting the origins of r_i and r_o and positioned such that the auxilliary clearance Sd/4 exists at both the inner and outer raceways. The auxilliary angle is

$$\alpha_o = \sin^{-1}\left[\frac{(r_i - D/2)\sin\alpha_s}{A}\right] \quad (3.7)$$

The angle associated with each ball bearing must be specified with the correct sign. A positive contact angle allows the bearing to accept a positively directed axial load transmitted by the shaft.

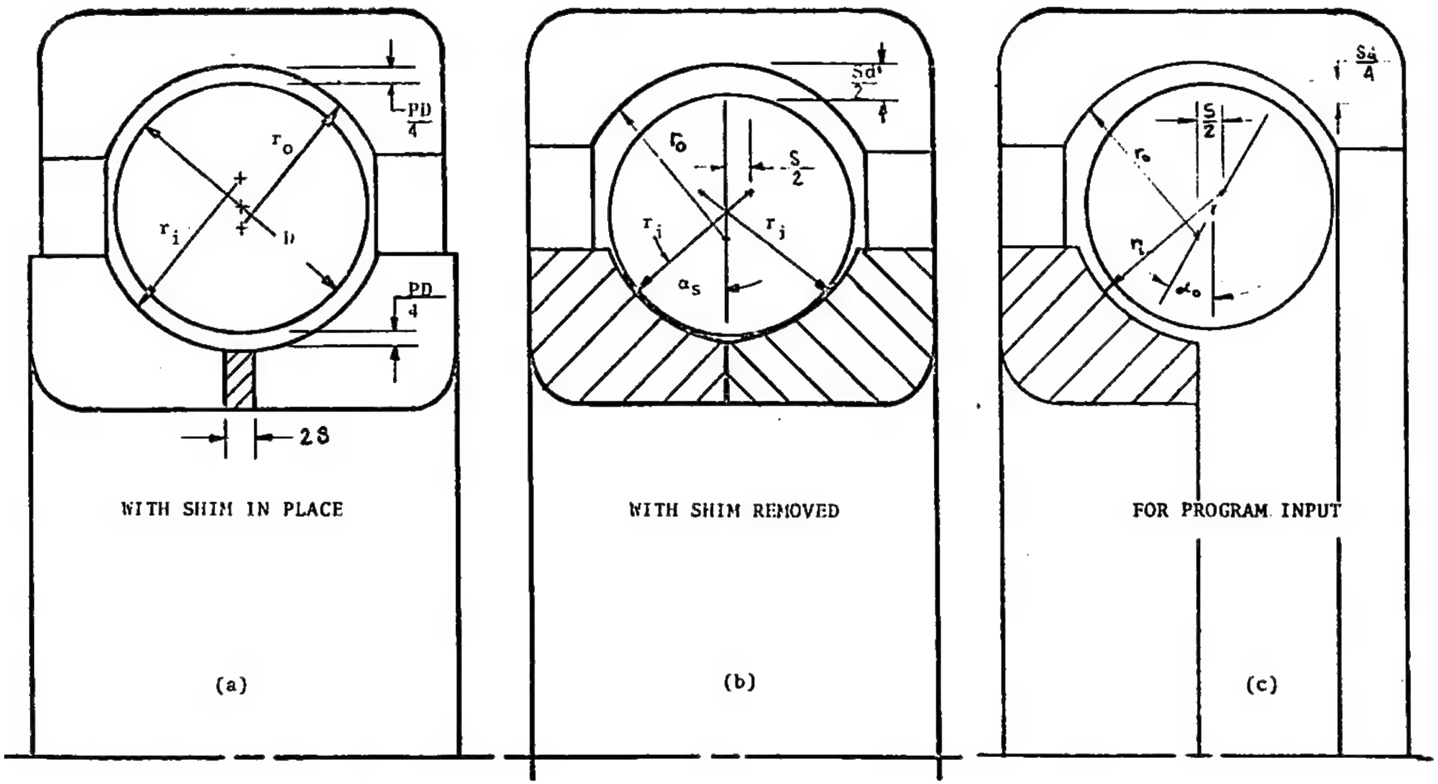


Figure 3.2 Split Inner Ring Ball Bearing Geometry

3.3.2.2 Tapered Roller Geometry

Items 2 and 6 are self-explanatory.

Item 1: Bearing Pitch Diameter

The tapered roller bearing pitch diameter may be calculated through specification of the roller measured large end diameter, the roller large end corner radius, one half the cup included angle, the roller included angle and the roller total length. This calculation is performed within the program. The user therefore need not specify the tapered roller bearing pitch diameter.

Item 3: Bearing Axial Play

For a tapered roller bearing, the bearing axial play rather than diametral clearance must be specified. (See Fig. 3.3) Note that this end play pertains to only the bearing in question. For two identical tapered roller bearings on a shaft, one half the total shaft axial play should be specified for each bearing. For two tapered roller bearings of dissimilar size, the total shaft axial play should be apportioned according to the bearing size such that the sum of the axial plays specified for the two bearings equals the total for the shaft.

Item 4: Bearing Contact Angle

For the tapered roller bearing, one-half the included cup angle (α_o) is input as the contact angle. (See Fig. 3.3) This angle must be specified with the correct sign. A positive angle allows the bearing to accept a positively directed axial load transmitted by the shaft and vice versa for a negative angle.

Item 5: Tapered Roller Bearing Flange Angle

The flange angle is shown by α_f in Fig. 3.3 The flange angle must always be positive.

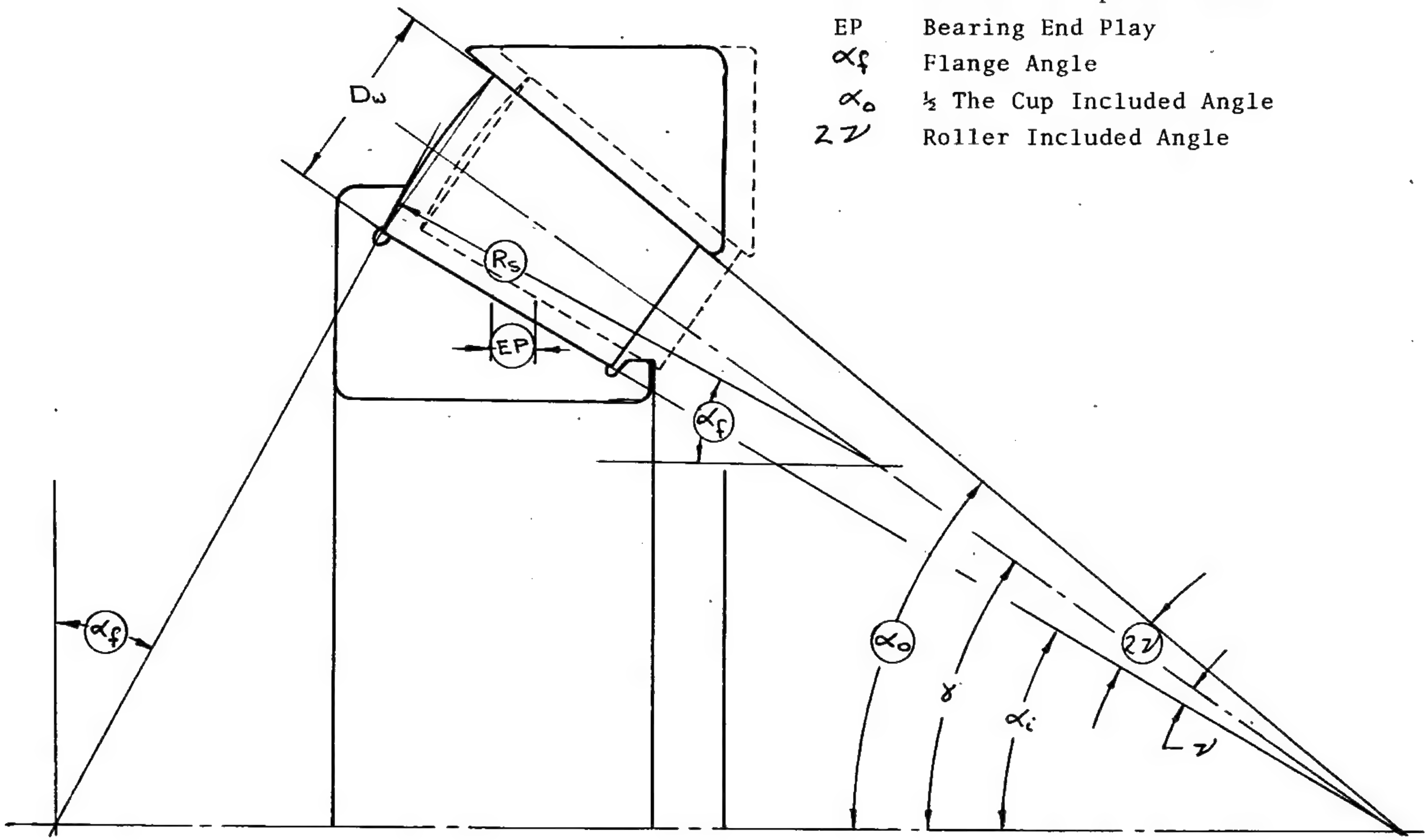
3.3.2.3 Cylindrical Roller Bearing Geometry

Items 1, 2, 3 and 6 are self explanatory. Both items 4 and 5 should be left blank.

FIG. 3.3 TAPERED ROLLER BEARING GEOMETRY

Note: Circled nomenclature represent input variable.

RS	Roller End Sphere Radius
EP	Bearing End Play
α_f	Flange Angle
α_o	½ The Cup Included Angle
2ν	Roller Included Angle



3.3.3 Card Type B3 - Rolling Element Geometry

3.3.3.1 Ball Geometry

The geometry of a ball is fully defined by its diameter.

3.3.3.2 Tapered Roller Geometry

Item 1 - Roller, Measured. Large End Diameter

For a tapered roller, the input diameter required is the largest measureable diameter. (See Fig. 3.4.) Typically this measurement should be taken where the large end corner radius becomes tangent to the roller surface profile. Within the program a "working" large end diameter is calculated. This diameter is shown as D_W in Fig. 3.4. All bearing geometrical relationships are calculated based on D_W.

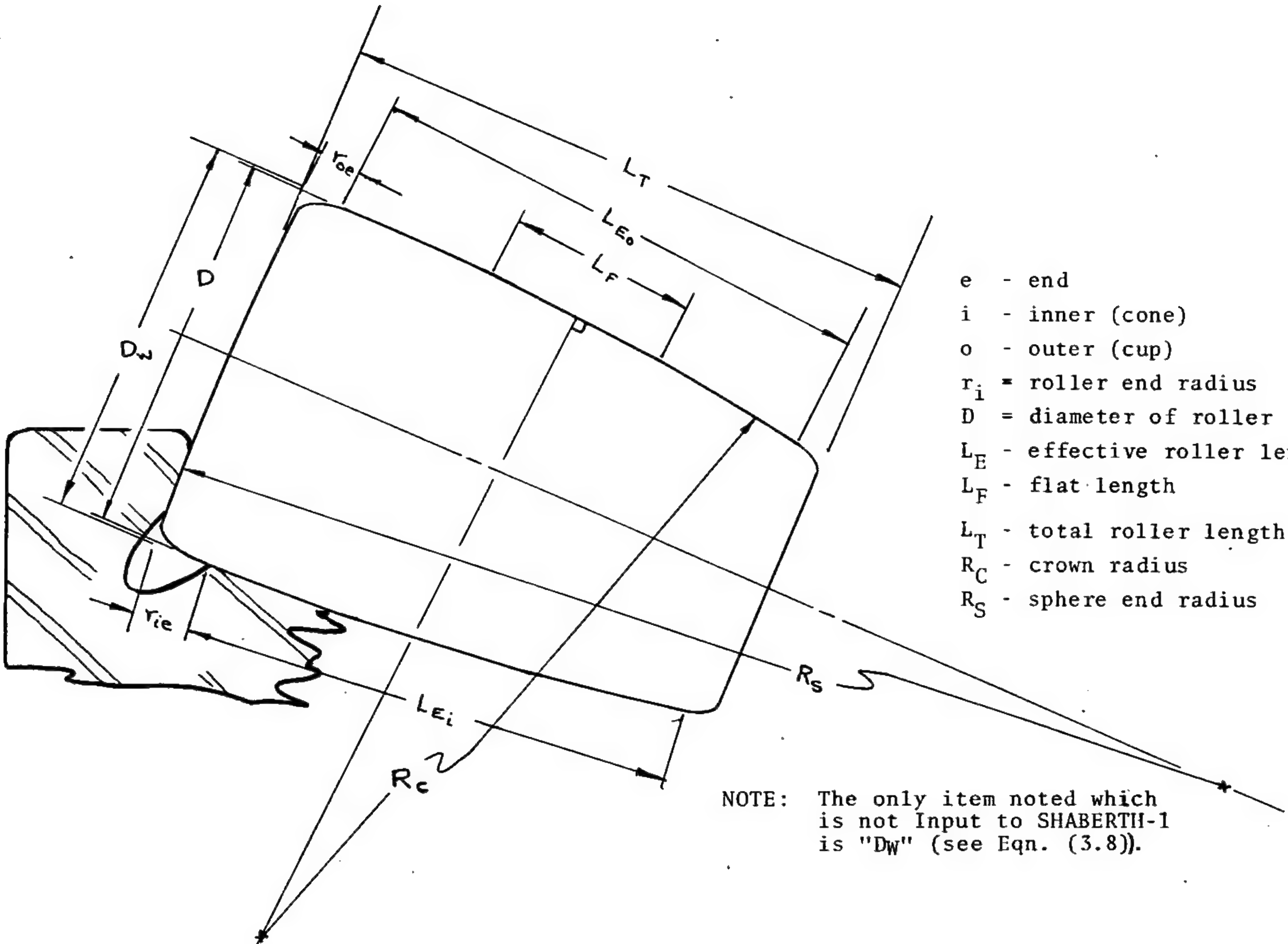
$$D_W = D + 2 r_{eo} \sin \nu \tag{3.8}$$

where: D is the measured large end diameter r_{eo} is the distance from the roller end to the beginning of the roller effective length at the outer raceway surface, measured parallel to the roller surface. If r_o is the corner radius at the roller large end

$$r_{eo} = \frac{r_o (1 + \sin \nu)}{\cos \nu} \tag{3.9}$$

ν is one half the roller included angle.

Item 2 through 6 are shown in Fig. 3.4 and are self explanatory. Note that the program can handle a nonzero roller flat length, Item 6. Most tapered rollers are, however, fully crowned.



e - end
i - inner (cone)
o - outer (cup)
r_i = roller end radius
D = diameter of roller
L_E - effective roller length
L_F - flat length
L_T - total roller length
R_C - crown radius
R_S - sphere end radius

NOTE: The only item noted which is not Input to SHABERTH-1 is "DW" (see Eqn. (3.8)).

FIG. 3.4 TAPERED ROLLER AND ROLLER-RACEWAY INPUT GEOMETRY

3.3.3.3 Cylindrical Roller Geometry

Typically, cylindrical rollers are partially crowned as shown in Fig. 3.5. The center of the roller is flat. Toward the ends the roller profile is formed by a crown radius. There are usually rounded corners at the roller ends. These corners reduce the load carrying surface of the roller such that if there are no raceway undercuts the roller raceway effective length equals the roller total length less the two corner radii (see Section 3.3.4.2). Note that a partially crowned roller is specified through input of a non-zero flat length. If the flat length is zero, the roller is fully crowned with its profile defined by the roller crown radius.

The roller end sphere radius is defined in Fig. 3.5. For a cylindrical roller, the roller included angle is zero.

3.3.4 Card Type B4 - Rolling Element-Ring Geometry

3.3.4.1 Ball Bearing

Items 1 and 2 refer to the outer and inner raceway curvatures respectively where curvature is defined as the cross groove radius divided by the ball diameter. Typical values range from 0.515 to 0.57.

3.3.4.2 Tapered Roller Bearing Contact Geometry

Items 1 and 2 - Roller-Raceway Effective Length

The roller-raceway load bearing surface is measured parallel to the roller surface such that if there were no relief at the roller ends the effective contact length would be:

$$L_e^* = L/\cos\nu \tag{3.10}$$

However, since the roller has corners at the large and small ends, the actual effective length is less than L_e^*. The consideration of raceway undercuts at the inner raceway flanges may result in a raceway effective length less than the roller effective length in which case the shorter of the two should be input.

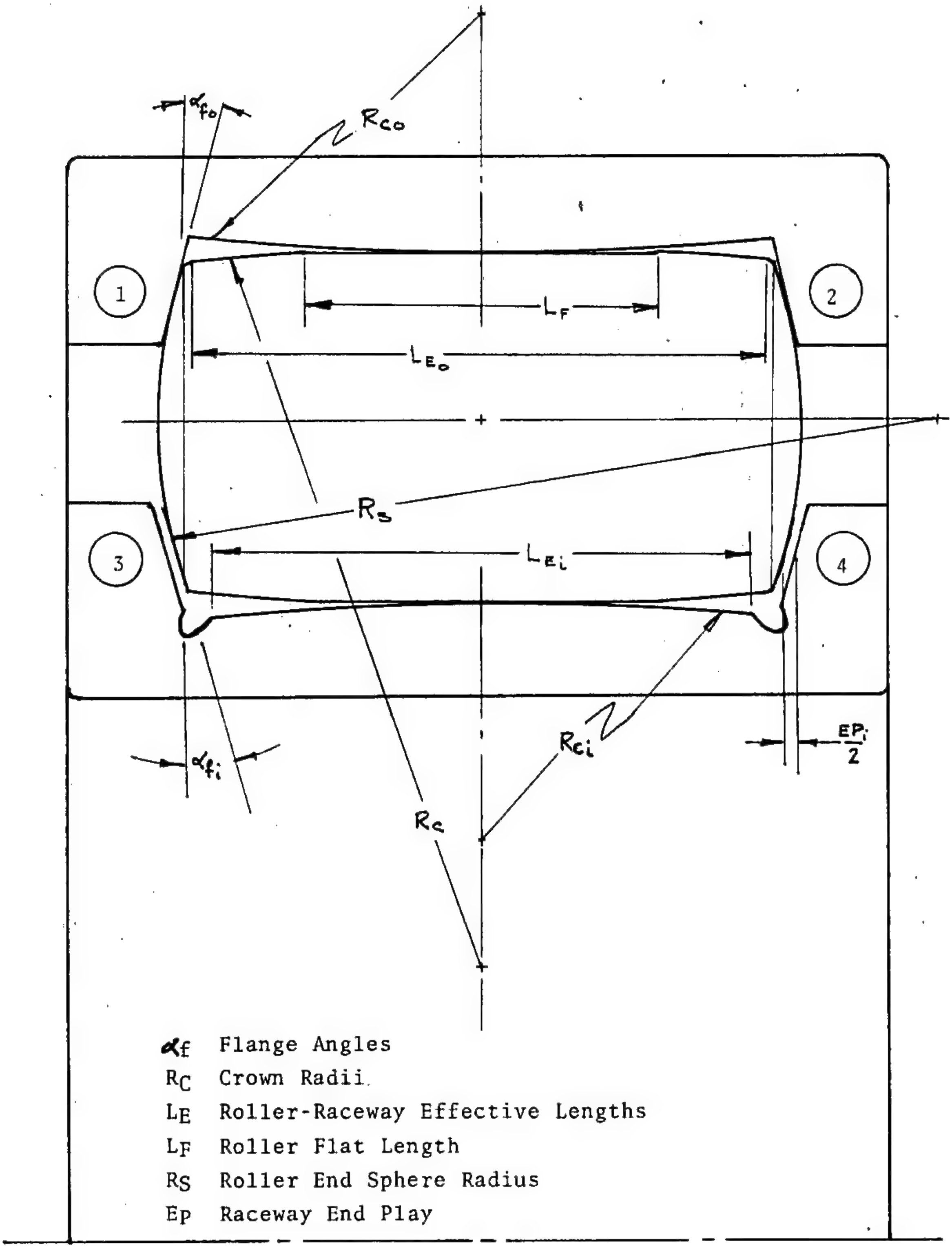


FIG. 3.5 CYLINDRICAL ROLLER BEARING GEOMETRY

Items 3 and 4 - Raceway Crown Radius

The present analysis permits both the roller and raceway to be crowned. If the raceway is crowned, it must be fully crowned with no flat specified. If the raceways are flat the input crown radius may be left blank, in which case a default value of 1. x 10^{+10} inches is used.

Note that the unloaded roller-raceway separation along the roller profile, (δc) Fig. 3.6, calculated at the center of each roller raceway slice is comprised of the sum of the roller and raceway crown drops.

Items 5 and 6 - Roller Large End Corner Relief

These data specify the distance from the roller large end to the point on the roller surface where the roller effective length begins. For the outer raceway contact this distance may be calculated using Eq. 3.9. For the inner raceway use Eq. 3.9 or the width of the inner raceway undercut at the large end.

Item 7 - The number of slices into which the roller raceway contacts are divided.

A maximum value of twenty (20) is permitted a default value of eleven (11) is used if a blank or zero is read.

3.3.4.3. Cylindrical Roller Contact Geometry

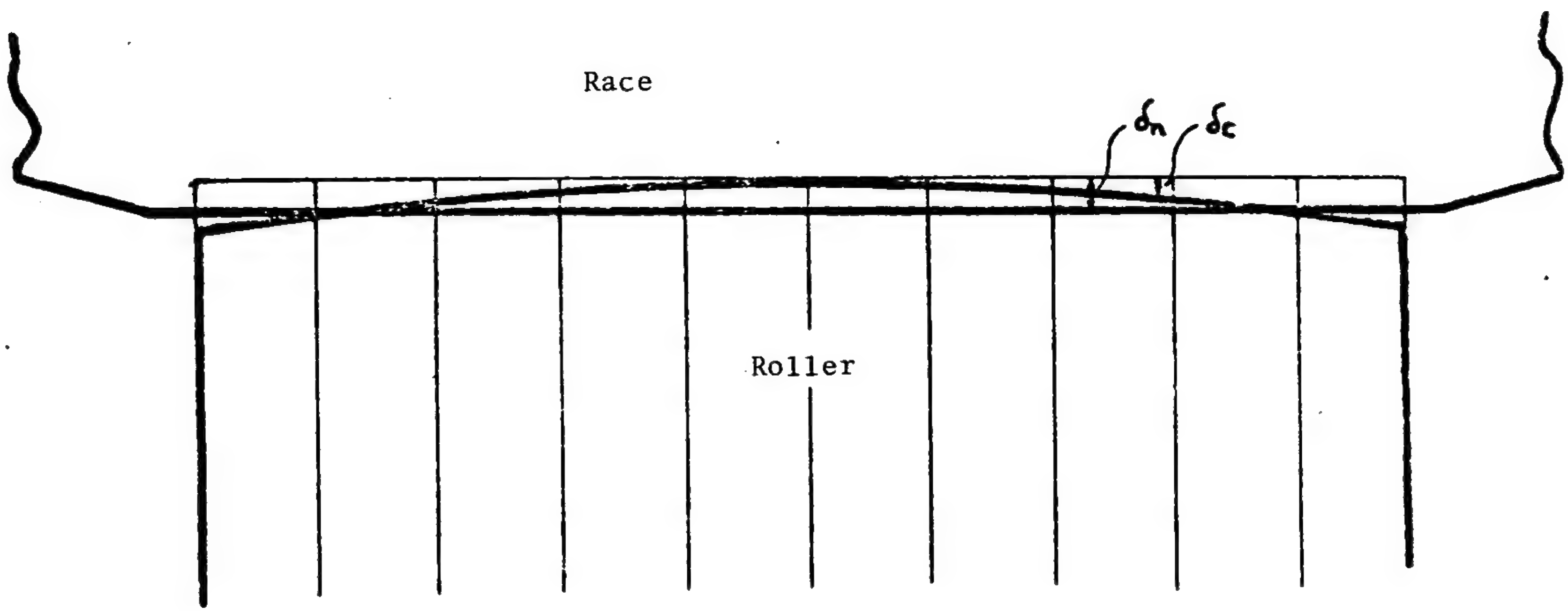
3.3.4.3.1 Card Type B4A Roler-Raceway Geometry

Items 1 through 4 and 7 have the same definitions as they had for the tapered roller bearing.

Items 5 and 6 - Roller End Corner Relief is not required input for the cylindrical roller bearing since the roller raceway effective length is assumed to be centered along the roller. This is not the case for the tapered roller bearing.

3.3.4.3.2 Card Type B4B Roller Flange Geometry for Cylindrical Roller Bearings

Figure 3.6 Roller-Race Lamination Showing Relative Approach (δu) and Crown Drop (δc)



Items 1 - 4 - The flange angle and end play definitions can be seen in Fig. 3.5. Note that when a ring has a single flange such as Inversion No. 3. Fig. 3.7, the end play is the distance between the roller and the flange when the roller is centered on the raceway.

Item 5 - The Flange Inversion Index - See Fig. 3.7 The number which corresponds to the particular Flange inversion being examined must be input. Note that since the inversions greater than eight (8) cannot carry axial loading, the bearing carries load only on the raceways, and thus the Program resets the inversion index to one if the input value is greater than eight. The inversion index must be input as a real number, (with a decimal point).

3.3.5 Roller-Raceway Non-Uniform Profile Definition

3.3.5.1 Card Type B5 - Inner Roller Raceway Contact

These cards are used to input the inner and outer race roller-race separation along the roller profile. With the high points of the roller and race in contact, i.e., with all clearance between roller and raceway removed. These cards must be omitted if item 7 of the Bearing Data Title card is zero or blank. These data are used only when the roller-raceway profile geometry cannot be defined by card types B3 and B4A.

3.3.6 Card Type B6 - Outer Roller Raceway Contact

Same as Card Type B5.

3.3.7 Ring-Rolling Element Surface Data

3.3.7.1 Card Type B7A - Raceway - Rolling Element Surface Data

Items 1 through 6 define the statistical surface micro-geometry parameters of the rollers and raceways. Items 1 through 3 require the input of center line average CLA surface roughness. Within the program CLA values are converted to RMS by multiplying by 0.9.

Items 4 through 6 are RMS values of the slopes measured in degrees, of the surface asperities as measured in a traverse across the groove for rings, longitudinally for rollers and in any arbitrary direction for balls. Typical values for raceway and rolling element surfaces are 1 to 2 degrees. This card is omitted if the solution level is NPASS = 0.

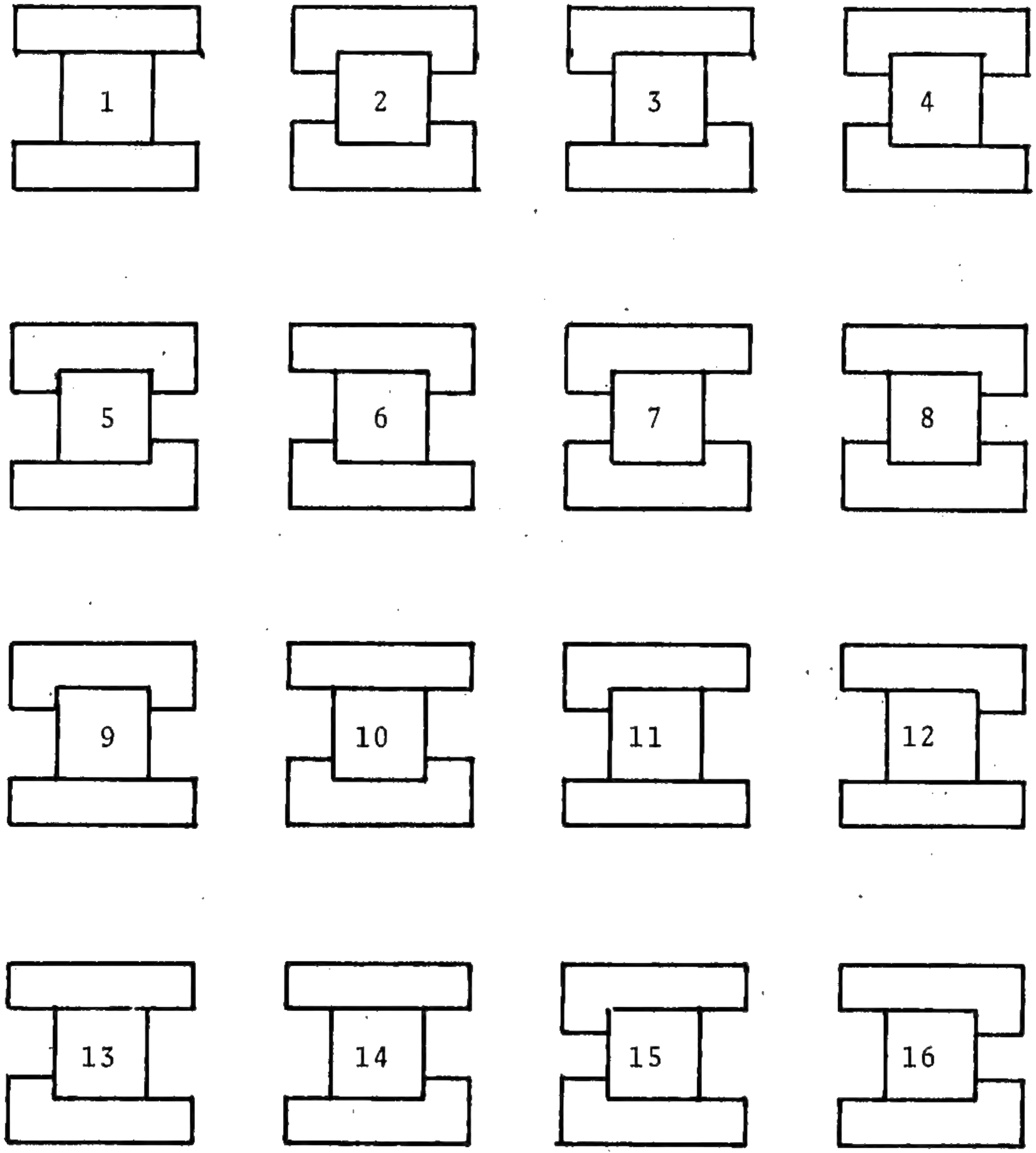


FIG. 3.7 CYLINDRICAL ROLLER BEARING FLANGE INVERSIONS

3.3.7.2 Card Type B7B - Flange-Roller End Surface Data

The data are identical to the data in Card Type B7A but refer to the flange and roller end surfaces rather than the raceway and roller rolling surfaces.

Note that both card types B7A and B7B are omitted if the solution level is NPASS = 0.

3.3.8 Card Type B8 - Cage Data

This card is omitted if the solution level is NPASS =0. These data are self explanatory. Note that the cage weight is an input item. The weight however, is not used in any calculation. It is included only for future consideration of cage stability predictions.

3.3.9 Card Type B9 - Shaft and Housing Fit Dimensions

These cards are to be included only if the change in bearing diametral clearance with operating conditions is to be calculated, i.e. if item 4,ITFIT, on the Bearing Title Card is non-zero. On Card Type B9, tight interference fits bear a positive sign and loose fits, a negative sign.

Item 3 and 6 on Card No. 9 are termed the shaft and housing effective widths respectively. The value specified for these effective widths may be as great as twice the ring width.

Use of an effective width is an attempt to account for the greater radial rigidity of a shaft being longer than the ring that is pressed on to it, owing to the fact that the shaft deflects over a distance that extends beyond the ring width. In the program the calculated internal pressure on the ring due to its interference fit with the shaft, is distributed over the shaft effective width and this (lower) pressure is used in computing the shaft deflection. Using double the actual width as the effective width is customary.

3.3.10 Card Type B10 - Shaft Housing Fit Dimensions

These items are self explanatory.

Note: Bearing System Components Material Properties. Card types B11 through B14.

These card types define the material properties of the shaft, inner ring, rolling element, outer ring and housing, data items 1 through 5 respectively. This set of cards is to be included if either the bearing clearance change analysis is used, i.e. item 4, ITFIT, Bearing Title Card 2 is non zero, or if the bearing rings or rolling elements are not steel, i.e. item 9, IMT, Bearing Title Card 2 is equal to 1. If any item on card types B11 through B14 is left blank, the program inserts the appropriate value of the steel property.

3.3.11 Card Type B11 - Elastic-Moduli

This card defines the elastic modulus for the shaft, inner ring, rolling element, outer ring, and housing respectively.

3.3.12 Card Type B12 - Poisson's Ratio

This card defines the Poisson's ratio for the shaft, inner ring, rolling element, outer ring, and housing respectively.

3.3.13 Card Type B13 - Density

This card defines the density for the shaft, inner ring, rolling element, outer ring, and housing respectively.

3.3.14 Card Type B14 - Coefficient of Thermal Expansion

This card defines the coefficient of thermal expansion for the shaft, inner ring, rolling element, outer ring, and housing respectively.

3.3.15 Card Type B15 - Lubrication and Friction Data

This card is omitted if the solution levels NPASS = 0.

Items 1 and 2

Items 1 and 2 are the amounts by which the combined thickness of the lubricant film on the rolling track and rolling element is increased during the time interval between the passage of successive rolling elements, from whatever replenishment mechanisms are operative. Item 1 applies to the outer and Item 2 to the inner race-rolling contacts respectively. If Item 1 is zero or blank the mode of friction is assumed to be dry.

At the present time the magnitude of the inner and outer raceway replenishment layers has not been correlated to lubricant flow rate, lubricant application methods and bearing size and speed factors. At this point then, the user is forced to establish proper values for the replenishment layer thickness. As a rough guide the following suggestions are made.

1) To avoid starvation, the replenishment layer thicknesses should be one to two times the EHD film thickness which develops in the rolling element raceway contacts.

2) Because of centrifugal force effects, intuition suggests that the outer raceway replenishment layer should be several times thicker than that prescribed at the inner raceway.

Item 3, XCAV, describes the percentage of the bearing cavity, estimated by the user to be occupied by the lubricant. $0 \leq XCAV \leq 100$.

As with the replenishment layer thicknesses, the amount of free lubricant should be able to be correlated with lubricant flow rate, lubricant application methods and bearing size and speed factors. At this time such correlations do not exist. XCAV values less than five percent are recommended at this point.

Item 4 is the coefficient of coulomb friction applicable for the contact of asperities. If Item 1 and 2 are zero, then Item 4 serves as the coulomb friction coefficient which prevails in all contacts.

Items 5 and 6 are the lubricant replenishment layer thicknesses for the outer and inner ring flanges respectively.

Item 7 is the coefficient of coulomb friction applicable for the asperity interactions at the roller end-flange contacts.

3.3.16 Card Type B16

This card is omitted if Item 9 title card 2 is zero or blank or if Item 1 card 15 is zero or blank which implies dry friction.

This card specifies the lubricant type. If Item 1, NCODE is 1, 2, 3, or 4 the Program uses preprogrammed lubricant properties as presented in Table 1, and no further information is required.

NCODE	Lubricant
1	A specific mineral oil
2	A MIL-L-7808G
3	Polyphenyl-Ether
4	A MIL-L-23699

NCODE may also be specified as negative (-1 to -4), in which case the traction characteristics of the respective lubricant NCODE noted above are used but the actual properties specified by Items 2 through 7 override the hard coded data. This option is most useful in specifying various mineral oils i.e. NCODE = -1.

3.4 DATA SET III - THERMAL MODEL DATA

Appendix B has been included to aid the user in data preparation and calculation of heat transfer coefficients required at input.

3.4.1 Card Type T1

Card type 1 is a control card. If no temperature map is to be calculated, this card is to be included as a blank card followed by a Type 2 card for each bearing on the shaft. Card Type 1 containes control input for both steady state and transient thermal analyses. It is not intended however, that both analyses be executed with the same run.

Item 1: The highest node number (M). The temperature nodes must be numbered consecutively from one (1) to the highest node number. The highest node number must not exceed one hundred (100).

Item 2: Node Number of the Highest Unknown Temperature Node (N). This number should equal the total number of unknown node temperatures. It is required that all nodes with unknown temperatures be assigned the lowest node numbers. The nodes which have known temperatures are assigned the highest numbers.

Item 3: Common Initial Temperature (TEMP)°C: The temperature solution iteration scheme requires a starting point, i.e., guesses of the equilibrium tempratures. Card Type 3 allows the user to input estimates of individual node temperatures. When a node is not given a specific initial temperature, the temperature specified as Item 3 of Card Type 1 is assigned.

Item 4: Punch Flag (IPUNCH): If the Punch Flag is not zero (0) or blank, the system equilibrium temperatures along with the respective node numbers will be punched according to the format of Card T3. This option is useful if, for instance, the user makes a steady state run with lubrication, and then wishes to use the resultant temperature as the initiation point for a transient dry friction run in order to assess the consequence of lubricant flow termination.

TABLE 1-A

LUBRICANT PROPERTIES OF FOUR OILS USED

Oil No.	Oil Type	Kinematic Viscosity, (cs) 100°F	210°F	Walther Equation Constants A	B	Density @ 60°F gm/cm^3 ρ	Thermal Conductivity Kf Btu/Hr/Ft/°R	Thermal Coeff. of Expansion G	Temp. Viscosity Coeff. β (°R^{-1})
1	Mineral Oil	64	8.0	10.349	3.673	0.8800	0.0671	3.52×10^{-4}	0.0193
2	MIL-L-7808G	12.8	3.2	10.215	3.698	0.9526	0.0879	3.94×10^{-4}	0.0132
3	C-Ether	25.4	4.13	11.452	4.113	1.201	0.0690	4.15×10^{-4}	0.0168
4	MIL-L-23699	28.0	5.1	10.207	3.655	1.010	0.0879	4.14×10^{-4}	0.0161

TABLE 1-B

TABULATION OF CONSTANT FOR FOUR OILS

Oil No.	Oil Type	λ	A_1	A_2	$p_1 \times 10^5$	C_1	C_2	$(\mu_r)_B$	$(V_s)_B$
1	Mineral (1) Oil	0.3	3.42	2.14	1.5	1.8	115.9	0.65	0.10
2	MIL-L-7808 (2)	0.94	4.08	1.48	1.7	17.6	39.0	0.68	0.25
3	C-Ether	0.70	3.29	1.50	1.7	26.2	6.6	0.65	0.15
4	MIL-L-23699 (2)	0.93	6.88	3.44	2.2	10.4	47.3	0.68	0.25

NOTES:

(1) obtained from Traction Data reported by Johnson and Cameron (28)

(2) S K F tests

(3) obtained from Traction Data reported by Smith et al (29)

Item 5: "Output Flag" (IUB). If the "Output Flag" is not zero, the bearing program output and a temperature map will be printed after each call to the shaft bearing solution scheme. This printout will allow the user to observe the flow of the solution and to note the interactive effects of system temperatures and bearing heat generation rates. Since the temperature solution is not mathematically coupled to the bearing solution the possibility exists that the solution may diverge or oscillate. In such a case, study of the intermediate output produced by the "Output Flag" option may provide the user with better initial temperature estimates that will effect a steady state solution. Two levels of bearing output are permitted. If IUB is 1, the rolling element output is not printed. If IUB is 2, full bearing output is obtained.

Item 6: "Maximum Number of Calls to the Shaft Bearing Program" (IT1). IT1 is the limit on the number of Thermal-Shaft-Bearing iterations, i.e., the external temperature equilibrium calculation. The user must input a non-zero integer such as 5 or 10 in order for the program to iterate to an equilibrium condition. If IT1 is left blank or set to zero (0) or 1, shaft bearing performance will be based on the initially estimated temperatures of the system. The temperatures printed out will be based on the bearing generated heats. It is unlikely that an acceptable equilibrium condition will be achieved. However, the temperatures which result may provide better initial estimates for a subsequent run than those specified by the user.

IT1 also serves as a limit on the transient temperature solution scheme, by limiting the number of times the shaft-bearing solution scheme is called. Each call to the shaft-bearing scheme will input a new set of bearing heats to the transient temperature scheme until a steady state condition is approached or until the transient solution time limit is reached.

Item 7: "Absolute Accuracy of Temperatures for the External Thermal Solution" (EPI). In the steady state thermal solution scheme, each calculation of system temperatures occurs after a call to the shaft-bearing scheme which produces bearing generated heats. After the system temperatures have been calculated for each iteration, using the internal temperature solution scheme, each node temperature is checked against the nodal temperature at the previous iteration.

If $\{t_{(N)i} - t_{(N-1)i}\} \leq$ EP1 for all nodes i then equilibrium has been achieved and the iteration process stops.

Item 8: "Iteration Limit for the Internal Thermal Solution" (IT2). After each call to the shaft bearing program, the internal temperature calculation scheme is used to determine the steady state equilibrium temperatures based on the calculated set of bearing heat generation rates. If the program is used to calculate the temperature distribution of a non bearing system it is the internal temperature scheme which is employed. If IT2 is left blank or set to zero, the number of internal iterations is limited to twenty (20).

Item 9: "Accuracy for Internal Thermal Solution" (EP2). The use of EP2 is explained in Section 3.6. If EP2 is left blank or set to zero (0), a default value of 0.001 is used.

Item 10: "Starting Time" (START) is a time in seconds T_s at which the transient solution begins; usually set to zero (0).

Item 11: "Stopping Time" (STOP) is the time in seconds at which the transient solution terminates, T_f. The transient solution will generate a history of the system performance which will encompass a total elapsed time of

$$(T_f - T_s) \text{ seconds}$$

Item 12: "Calculation Time Step" (STEPIN). The transient internal solution scheme solves the system of equations

$$t_{k+1} = t_k + \frac{q_k}{\rho C_p V} \Delta T \qquad (3.11)$$

$$\Delta T = \text{STEPIN}$$

The user may specify STEPIN. If left blank or set to zero (0), the Program calculates an appropriate value for STEPIN using the procedure described in Section 3.6.

Item 13: "Time Interval Between Printed Temperature Maps" (TTIME) seconds. The user must specify the length of time in seconds which will elapse between each printing of the temperature map. The interval will always be at least as large as the "calculation timestep" (STEPIN).

Item 14: "Time Interval Between Calls of the Shaft Bearing Portion of the Program" (BTIME). BTIME will always have a value larger than or equal to (STEPIN) even if the user inadvertently inputs a shorter interval. Computational time savings result if BTIME is greater than STEPIN, however, accuracy might be lost.

3.4.2 Card Type T2

Card Type T2 is required, one card for each bearing if no thermal analysis is being performed. The temperature data is used within the shaft-bearing analysis portion of the program to fix temperature dependent properties of the lubricant in which case the inner race, outer race, lubricant bulk cavity and flange temperatures are used. Note the flange numbering scheme depicted in Fig. 3.5. The tapered roller bearing, inner ring flange is considered flange No.1. The assembly component temperatures at each bearing location are used in the analysis which calculates the change in bearing diametral clearance from "off the shelf" to operating conditions.

3.4.3 Card Type T3

In the steady state analysis this card is used to input initial guesses of individual nodal temperatures for unknown nodes as well as the constant temperatures for known nodes, such as ambient air and/or an oil sump.

In the transient analysis, Card Type T3 is used to input the nodal temperatures of all nodes at (START) = T_S i.e. at the initiation of the transient solution.

3.4.4 Card Type T4

With this card, node numbers are assigned to the components of each bearing, one card per bearing. With this information the proper system temperatures are carried into each respective bearing analysis. The inner race and inner ring node numbers may or may not be the same at the user's discretion. Similarly the outer race and outer ring node numbers may or may not be the same.

3.4.5 Card Type T5

The shaft bearing system analysis accounts for frictional heat generated at four locations in the bearing, i.e. at the inner race, the outer race, between the cage rail and ring land, and in the bulk lubricant due to drag. The heat generated at the hydrodynamic cage-rolling element contact is added to the bulk lubricant. Heat generated at the flange is not presently considered. This card allows the heat generated to be distributed equally to two nodes. For instance the heat generated at the inner race-rolling element contact should be

distributed half to the rolling element and half to the inner race. The heat developed between the cage and inner ring land may be distributed half to the inner ring and half to the cage if a cage node has been defined, otherwise, half to the bulk lubricant.

3.4.6 Card Type T6

This card specifies the node numbers and the heat generation rate for those nodes where heat is generated at a constant rate such as at rubbing seals or gear contacts.

3.4.7 Card Type T7

This card type is used to input the numerical values of the various heat transfer coefficients which appear in the equations for heat transfer by conductivity, free convection, forced convection, radiation and fluid flow. Up to ten coefficients of each type may be used. Separate values of each type of coefficient are assigned an index number via card T7 and in describing heat flow paths (Card Type T8 below) it is necessary only to list the index number by which heat transfers between node pairs.

Indices 1-10 are reserved for the conduction coefficient λ, 11-20 for the free convection parameters, 21-30 for forced convection, 31-40 for emissivity and 41-50 for fluid flow (product of specific heat, density and volume flow rate).

As an example, for heat transfer by conduction with coefficient λ of 53.7 watts/M°C one could prepare a card type T7 with the digit 1 punched in column 10 and the value 53.7 punched in the field corresponding to card columns 11-20. If a conduction coefficient of 46.7 were applicable for certain other nodes in the system one could punch an additional card assigning index No. 2 to the value λ = 46.7 by punching a "2" in card column 10 and 46.7 anywhere within card columns 11-20.

Rather than inputting constant forced convection coefficients, optionally, these coefficients can be calculated by the program in one of three ways. If the calculation option is exercised a pair of cards is used in place of a single card containing a fixed value of α. The contents of the pair of cards depends upon which of the three optional methods are used.

Option 1) α is independent of temperature but is calculated as a function of the Nusselt number which in turn is a function of the Reynolds number Re, the Prandtl number P_r as follows, (cf. {17})

$$\alpha = N_u \lambda_{oil}/L$$

$$N_u = aR_e^b P_r^c$$

where λ_{oil} is the lubricant conductivity, L is a characteristic length (with a unit of meters) and c, a and b are constants.

Option 2) α is a function only of fluid dynamic viscosity and viscosity is temperature dependent.

$$\alpha = c\eta^d$$

Option 3) α is a function of the Nusselt, Reynolds and Prandtl numbers and viscosity is temperature dependent.

3.4.8 Card Type T8

This card defines the heat flow paths between pairs of nodes. Every node must be connected to at least one other node, i.e., two or more independent node systems may not be solved with a single Program execution.

The calculation of heat transfer areas is based on lengths, L_1 and L_2 input using Card Type T8. Additionally, the type of surface for which the area is being calculated is indicated by the sign assigned to the heat transfer coefficient index. If the surface is cylindrical or circular the index should be positive, if the surface is rectangular the index should be input as a negative integer.

In the case of radiation between concentric axially symmetric bodies, L_3 is the radius of the larger body. For radiation between two parallel flat surfaces or for conduction between nodes, L_3 is the distance between them.

Fluid flow heat transfer accounts for the energy which the fluid transports across a node boundary. Along a fluid node at which convection is taking place, the temperature varies. The nodal temperature which is output is the average of the fluid temperature at the output and input boundaries. If the emerging temperature of the fluid is of interest, it is necessary to have a fluid node at the fluid outlet. At this auxiliary node only fluid flow heat transfer occurs and the fluid temperature would be constant throughout the node. Thus the true fluid outlet temperature will be obtained.

Conduction of heat through a bearing is controlled by index 51. The actual heat transfer coefficient which contains a conductivity, area and a path length term is calculated in the bearing portion of the program. The term is based upon conduction through an average outer race and inner race rolling element contact.

3.4.9 Card Type T9

This card inputs data required to calculate the heat capacity of each node in the system. This card type is required only for a transient analysis.

3.5 DATA SET IV - SHAFT INPUT DATA

The shaft-bearing analysis requires all loading to be applied to the shaft. The loads applied to each bearing are a product of the shaft-bearing solution. There is no need for the user to solve the statically determinate or indeterminate system for bearing loads. Even if a single bearing is being analyzed, with the applied load acting through the center of the bearing, data for a dummy shaft must be supplied.

In the analysis the housing is assumed to be rigid. Provision has been allowed to input data for housing radial and angular spring characteristics. However, this has been done for future consideration of an elastic housing and is therefore currently unavailable.

The shaft input data consists of three card types:

1) Shaft Geometry and Elastic Modulus Data
2) Bearing Position and Mounting Error Data
3) Shaft Load Data

3.5.1 Card Type S1

This card type is used to describe shaft geometry at up to twenty locations along the shaft. The user must place his shaft in a cartesian coordinate system with the end of the shaft at the origin and with the shaft lying along the X-axis.

The shaft may have stepwise and linear diameter variations. The stepwise variations require a single card which specifies different diameters immediately to the left and right of the relevant X shaft coordinate. The shaft analysis assumes a linear diametral variation if on two successive cards, i.e. two successive X coordinates, the diameter to the right of the location differs from the diameter to the left of the location

of the following card. Complex shaft geometries may be approximated with a set of linear diameter variations spaced at close intervals.

If an Elastic Modulus is not specified at the designated input location, the modulus of steel is assumed, 204083N/mm^2.

3.5.2 Card Type S2

This card type locates the bearing inner ring on the shaft in the X-Y and X-Z planes. For a ball bearing, the X coordinate specified locates the inner ring center of curvature. For cylindrical roller bearings the X coordinate locates the center of the inner race roller path.

For tapered roller bearings in the strictest sense the X coordinate locates the point where a line from the roller center of gravity, intersects and is perpendicular to the inner raceway, with all bearing end play removed. It is sufficiently accurate however to allow the X coordinate to locate the center of the inner raceway.

In addition to specifying bearing location, the Type 2 card is also used to specify housing radial and angular mounting errors. As mentioned previously, space has been reserved for inputing housing radial and angular spring characteristics, however, these characteristics are not used in the system analysis.

Two sets of Type 2 cards may be required. The first set is always required and defines housing alignment errors in the shaft X-Y plane. The second set defines the housing alignment errors in the shaft X-Z plane and is required only if non zero errors exist for the particular bearing in question.

The first set of Type 2 cards must contain a card for each bearing.

3.5.3 Card Type S3

Type 3 cards are used to specify shaft loadings at a given X coordinate. Loading may be applied in the X-Y and X-Z planes, thus requiring two distinct sets of Type 3 cards. Applied loads may have the form of concentrated radial forces, concentrated moments, linearly distributed radial forces and concentrated axial loads which may be eccentrically applied. If an axial load is eccentrically applied, the moment which results must not be separately calculated and input as a concentrated moment.

Variations in distributed radial loads are handled at input just as shaft linear diameter variations are handled.

Note that each set of Type 3 cards must be followed by a blank card.

Also note that in order for symmetry conditions to be considered the second Type 3 card must be void of any loading data.

4.0 COMPUTER PROGRAM OUTPUT

4.1 Introduction

The Program Output is intended to provide the engineer or designer with a complete picture of the shaft-bearing system performance.

In addition to the calculated output data, the input data is listed, thus producing a complete record of the computer run.

A sample output is included in Appendix D. The multi-bearing supported shaft problem demonstrates the full capabilities of SHABERTH including the steady state thermal and bearing clearance change solutions.

Key output items are discussed briefly below.

4.2 Bearing Output

4.2.1.1 Linear and Angular Deflections

These deflections refer to the bearing inner ring relative to the outer ring and are defined in the inertial coordinate system of Figure 2.4. The bearing deflections are not necessarily equal to the shaft displacements since the bearing outer ring radial or angular mounting errors may be specified as non-zero input.

4.2.1.2 Reaction Forces and Moments

These values reflect bearing reactions to shaft applied loading and outer ring mounting errors.

When the bearing inner ring has achieved an equilibrium position, the summation of all bearing reaction loads should numerically equal the shaft applied loading. When the level of solution indicated by "NPASS" = 2 is employed, as discussed in Section 5, differences between shaft applied and bearing reaction loads will exist but will typically be less than 10%. This difference is a consequence of friction forces contributing to the reaction loads whereas the inner ring equilibrium position has been determined considering elastic contact forces only.

4.2.2 Fatigue Life Data

The L_{10} fatigue life of the outer and inner raceways as well as the bearing are presented. The bearing life represents the statistical combination of the two raceway lives. These lives reflect the combined effects of the lubricant film thickness and material life factors. The lubricant film thickness life factor is described in detail in Section 3.

4.2.2.1 h/sigma

The ratio h/σ, also referred to as λ, is printed for the most heavily loaded rolling element. The variable h, represents the EHD plateau film thickness with thermal and starvation effects considered. The variable σ, represents the composite root mean square surface roughness of the rolling element and the relevant raceway.

4.2.2.2 Life Multipliers

4.2.2.2.1 Lubrication - This life multiplier is a function of h/σ at each concentrated contact and is in the form of a derating factor. Its value ranges from 0.479 for $h/\sigma = 0$ to 1.0 at $h/\sigma \geq 4$. Since the lubricant life multiplier is decremental the normal multiple of 3 used for thick film lubrication must be multiplied by the material life factor normally used and this product should be specified at input. This subject is covered in more detail in Section 3.3.1.

4.2.2.2.2 Material - This output simply reflects the input value, Again, it is covered in Section 3.

4.2.3 Temperatures Relevant to Bearing Performance

These temperatures fully describe the temperature conditions which affect the performance of a given bearing. If one of the temperature mapping options is used, the temperatures printed reflect the results of the particular option. If, neither temperature option was used, the list is simply a repeat of the input data. Note that there are separate temperatures for outer and inner raceways and flanges and ring temperatures. The raceway and flange temperatures are used to determine lubricant properties. The ring temperatures are used in the bearing dimension change analysis. The raceway, flange and ring temperatures may be the same value.

4.2.4 Frictional Heat Generation Rate and Bearing Friction Torque

4.2.4.1 Frictional Heat Generation Rate

The various sources of frictional heat generated within the bearings are listed. The values printed for "OUTER RACE, OUTER RING FLANGES, INNER RACE, INNER RING FLANGES, R.E.DRAG AND R.E. CAGE" represent the sum of the generated heats for all rolling elements. Additionally, the heats printed for the outer and inner raceways and flanges, plus the rolling element cage, reflect the friction developed outside the concentrated contacts, i.e.,.the HD friction as well as the EHD friction developed within the concentrated contacts. The raceway and flange data also includes any heat generated as a consequence of asperity contacts. "R. E. DRAG" should be interpreted as the heat resulting from lubricant churning as the rolling elements plow through the air-oil mixture.

4.2.4.2 Torque

The torque value is calculated as a function of the total generated heat and the sum of the inner and outer ring rotational speeds. The intent is to present a realistic value of the torque required to drive the bearing. Under conditions of inner ring rotation the torque value reflects the torque required to drive the inner ring. Since the inner ring torque also includes that friction torque required to impart an angular velocity to the lubricant in the bearing. A considerable portion of the lubricant will come to rest within the housing and not at the outer ring. Thus the measured outer ring torque may not equal the torque at the inner ring.

4.2.5 EHD Film and Heat Transfer Data

4.2.5.1 EHD Film Thickness

These values refer to the calculated EHD plateau film thickness at both contacts of the most heavily loaded rolling element and include the effects of the thermal and starvation reduction factors.

4.2.5.2 Starvation Reduction Factor

These factors give for the inner and outer ring contacts, the reduction in EHD film thickness due to lubricant film starvation according to the methods of Chiu, (11).

These factors pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

4.2.5.3 Thermal Reduction Factor

These factors are calculated according to the methods of Cheng, (10) and pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

4.2.5.4 Meniscus Distance

These factors are calculated according to the methods of Chiu, (11) and pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded

rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

4.2.5.5 Raceway-Rolling Element Conductivity

These data reflect the amount of heat transfer between rolling element and raceway for each degree centigrade difference between the two components. These data reflect the average of all outer and inner contacts respectively.

4.2.6 Fit and Dimensional Change Data

4.2.6.1 Fit Pressures

These data refer to the pressures built up as a consequence of interference fits between shaft and inner ring and housing and outer ring. Pressures are presented both for the standard cold-static condition (16°C) and at operating conditions.

4.2.6.2 Speed Giving Zero Fit Pressure (Between the shaft and inner ring)

This is a calculated value based upon operating conditions and provides a measure of the adequacy of the initial shaft fit.

4.2.6.3 Clearances

"Original" refers to cold unmounted clearance which is specified at input if the diametral clearance change analysis is executed. "Change" refers to the change in diametral clearance at operating conditions relative to the cold unmounted condition. A minus sign indicates a decrease in clearance. "Operating" refers to the clearance at operating conditions. For all types of ball bearings the decrease in clearance can be combined with the initial diametral clearance, and the free operating contact angle at operating conditions may be calculated. Note that the change in clearance should be compared against the diametral play of the split inner ring ball bearing in order to determine if the possibility for three point contact exists. The Program does not account for three point contacts even though the change in clearance might suggest that three point contact is obtained.

4.2.7 Lubricant Temperatures and Physical Properties

The lubricant properties, particularly the dynamic viscosity and to a lesser degree the pressure viscosity coefficient, are heavily temperature dependent. These factors enter the EHD film thickness calculation and the HD and EHD friction models. The lubricant is assumed to be at the same temperature as the relevant raceway. As noted elsewhere, these temperatures may be either input directly or calculated by the Program.

The physical properties printed are self explanatory. The units are enumerated.

4.2.8 Cage Data

4.2.8.1 Cage-Land Interface

The cage data indicates the performance parameters at the interface between the cage rail and the ring land on which the cage is guided. The torque, heat rate and separating force require no explanation. The eccentricity ratio defines the degree to which the cage approaches the ring on which it is guided at the point of nearest approach. The radial displacement of the cage relative to the bearing axis is divided by one half the cage-land diametral clearance. An eccentricity ratio of one indicates cage-land contact. A ratio of zero indicates that the cage rotation is concentric with the bearing axis.

Only the cage-land and rolling element pocket forces are considered in determining the cage eccentricity. The cage weight and centrifugal force which result from the eccentricity although available, are not considered in the analysis. The omission of these considerations helps reduce convergence problems.

4.2.8.2 Cage Speed Data

Cage speed data presents the comparison between the cage speed calculated based upon the quasidynamic equilibrium considerations and the speeds calculated with raceway control theory for ball bearings and the epicyclic speeds of the roller bearing components.

4.3 Rolling Element Output

4.3.1 Rolling Element Kinematics

4.3.1.1 Rolling Element Speeds

All of the rolling element speeds tend to vary from position to position when the bearing is subjected to combined loading.

The total rolling element speed is with reference to the cage and represents the vector sum of the three orthogonal components.

4.3.1.2 Speed Vector Angles

The rolling element speed vector angles, Arctan (ω_y/ω_x) and Arctan (ω_z/ω_x) are presented in order to show a clearer picture of the predicted ball kinematics. The ball speed vector tends to become parallel with the bearing X axis with increasing shaft speed and decreasing contact friction.

4.3.2 Rolling Element Raceway Loading

4.3.2.1 Normal Forces

The normal forces acting on each rolling element are printed. The rolling element race normal forces are self explanatory. The cage force is calculated only when the friction solution is employed and is always directed along the rolling element Z axis. If the rolling element orbital speed is positive, a positive cage force indicates that the cage is pushing the rolling element, tending to accelerate it. Cage force is a function of rolling element position within the cage pocket. Its magnitude is derived using hydrodynamic lubrication assumptions, when the distance between the rolling element and cage web is large, and EHD assumptions when the separation is of the order of the EHD film thickness or when rolling element-cage web interference exists.

4.3.2.2 Hertz Stress

The stress printed represents the maximum normal stress at the center of each ball race contact or at the most heavily loaded slice of the roller raceway contact.

4.3.2.3 Load Ratio Q_{asp}/Q_{tot}

If the EHD film thickness is small compared to the RMS composite rolling element-race surface roughness, the rolling element-race normal load will be shared by the EHD film and asperity contacts. The load ratio reflects the portion of the total load carried by the asperities.

4.3.2.4 Contact Angles

A ball bearing, subject to axial loading, misalignment or mounted such that the inner ring is always displaced axially relative to the outer rings, (i.e. a duplex set of angular contact ball bearings) will have non-zero contact angles. At low ball orbital speeds the inner and outer race angles will be substantially the same. At high speeds, ball centrifugal force will cause the outer race contact angle to be less than the inner race angle.

4.3.3 Roller End-Flange Contact Data

For a tapered roller bearing a single set of roller end-flange data is printed. For the cylindrical roller bearing, which may have up to four flanges, the program examines the data and prints the results for the most heavily loaded outer and inner ring flanges. In addition to the data listed below for a cylindrical roller, the semimajor contact axis as well as the concentrated contact and hydrodynamic heat generation rates are printed for the tapered roller.

4.3.3.1 Normal Force

The interference between the roller end and flange is determined from the solution for the relative, ring and rolling element positions. Hertz theory is used to calculate the load which results from this interference.

4.3.3.2 Hertz Stress

The Hertz stress printed is the maximum normal stress which occurs in the contact.

4.3.3.3 EHD Film Thickness

The plateau film thickness is calculated using the Archard Cowking equation for point contacts. This result is then modified to account for starvation (11) and thermal (10) reduction effects. This modified film thickness is printed.

4.3.3.4 Sliding Velocity

The sliding velocity is defined as the difference between the flange and roller end linear velocities at the center of the contact.

4.3.3.5 Rolling Velocity

Rolling velocity is defined as one half the sum of the roller end and flange linear velocities at the center of the contact.

4.3.3.6 Contact Ellipse Semiminor Axis

To help assess the severity of the roller end flange contact and the possibility for edge loading, the semiminor axis of the contact is printed.

4.4 Thermal Data

As in the case for bearing output, all of the input data are printed. The calculated output data are presented in the form of a temperature map in which a node number and the respective node temperature appear. The appearance of the steady state and transient temperature maps are identical. The transient temperature map also includes the time (T) at which the temperature calculations were made.

4.5 Shaft Data

These data simply reflect the input information.

4.6 Program Error Messages

4.6.1 From Subroutine ALLT

"Steady State Solution with (EP1) degrees accuracy was not obtained after (IT1) Iterations".

This message pertains to the external temperature iteration scheme in which system temperatures and bearing generated heats are being solved for an equilibrium condition.

4.6.2 From Subroutine SHABE

"It was not possible to obtain the change of clearance with an accuracy of (ERFIT) times the rolling element diameter in (ITFIT) iterations".

This message pertains to the bearing diametral clearance change iteration scheme. The solution may be converging in which case the number of iterations (ITFIT) should be increased. This can be checked with an NPRINT = 1 intermediate printout. The intermediate print may indicate that the solution is oscillating. The most likely cause of oscillation is the alternate prediction of bearing preload with all rolling elements loaded, and then in the next iteration, only a subset of the rolling elements loaded. This problem can usually be overcome by either of two methods.

1) In subroutine FIT remove the GO TO 20 statement. This will cause the inner ring load distribution to have no effect on the change in diametral clearance.

2) The solution can be damped by redefining the solution damping factor FA, such that it would take on a value $0 \leq FA \leq 1$. FA is presently set to 1 in subroutine SHABE. If this damping technique is used, the number of FIT iterations should be increased as the value of FA is decreased. An upper limit of 10 iterations is recommended.

4.6.3 <u>From Subroutine SOLVXX</u>

1) "SINGULAR SET OF EQUATIONS"

   This message might occur when the thermal input data is not input properly.

2) "THE LIMIT FOR NUMBER OF ITERATIONS IS REACHED"

   This message might occur either during a steady state temperature solution or bearing solution. Before increasing the number of iterations check the equation residue values. If they are low, the solution may be good enough.

3) "THIS IS THE BEST WE CAN DO. IT MAY BE USEABLE"

   This message reflects the fact that the next iteration will result in divergence. The iteration procedure is thus terminated. The equation residue values should be checked. If they are low, the solution may be useable.

As noted above the occurrence of messages 2) and 3), do not necessarily mean that the solution is not good. Generally the messages indicate that the solution has not converged as tightly as the user has requested.

Note: The XX suffix on SOLV specifies the version of SOLV contained in the user's program. As of this writing the current version is SOLV13. The suffix is changed each time improvements are made which require a change in the SOLVXX calling sequence.

4.6.4 From Subroutine INTFIT

"Singular matrix on tight shaft fit"
"Singular matrix on loose shaft fit"

These messages reflect an error in the input data usually as a consequence of inconsistent component diameters, such as the shaft inside diameter being greater than the outside diameter.

5.0 GUIDES TO PROGRAM USE

The Computer Program is a tool. As with any tool the results obtained are at least partially dependent upon the skill of the user. Certainly the economics of the Program usage are highly dependent upon the user's technical need and discriminate use of Program options.

Some general guides for efficient use of the Program are listed below:

1. Attempt to use the lowest level of solution possible. For instance if the prime object of a given run is to obtain bearing fatigue lives, execute only the elastic solution (NPASS = 0). If an estimate of bearing frictional heat is required, execute the low level friction evaluation (NPASS = 1). Execute the friction solution (NPASS = 2) only if rolling element and cage kinematics are of interest. Execute the highest level of solution (NPASS = 3) if kinematics are of interest and the bearing reaction loads deviate substantially from the shaft applied loading, i.e. a deviation greater than ten percent.

2. Attempt to input bearing operating diametral clearance rather than calculate it. Or, execute the diametral clearance change analysis once for a group of similar runs and use the output from the first run as input to the subsequent runs, omitting the clearance change analysis.

3. Attempt to input accurate operating temperatures rather than calculate them.

4. The more non-linear the problem the more computer time required to solve it. In the bearing friction solution large coefficients of friction seem to increase the degree of nonlinearity. In the thermal solutions, if possible, eliminate nonlinearities by omitting radiation terms and by using constant rather than temperature dependent free and forced convection coefficients.

5. In the transient thermal solution, space the calls to the shaft-bearing solution (BTIME) to as large an interval as prudently possible. Be careful however, too long an interval will produce large errors in heat rate predictions.

6. In the steady state thermal analysis attempt to estimate nodal temperatures on a node by node basis. Nodes which are heat sources should have higher temperatures than the surrounding nodes.

The above suggestions are intended to encourage the use of the Program on a cost effective basis. The intent is not to discourage the use of important program capabilities but to emphasize how the program should be most effectively used.

It is suggested that the user take a simple, axially loaded ball bearing problem and execute the program through the full range of options beginning with a frictionless solution proceeding to the three levels of friction solution with a low (0.01) and high (0.1) friction coefficient. The diametral clearance change analysis and the thermal solutions should also be executed on an experimental basis. This exercise will provide the user with some insight into economics of the Program usage on his computer as well as the results obtained from various levels of solution of the same problem.

It is also suggested that a constant user of the program should study the hierarchical Program flow chart Appendix A, along with the Program listing to gain an appreciation of the program complexity and the flow of the problem solution. The Program is comprised of many small functional subroutines. Knowledge of these small elements may allow the user to more easily piece together the philosophy of the total problem solution.

LIST OF REFERENCES

1. Kellstrom, M., "A Computer Program for Elastic and Thermal Analysis of Shaft Bearing Systems", S K F Report No. AL74P014, submitted to Vulnerability Laboratory, U. S. Army Ballistic Research Laboratories, Aberdeen Proving Ground, MD, under Army Contract DAAD05-73-C-0011.

2. Harris, T., "An Analytical Method to Predict Skidding in High Speed Roller Bearing", ASLE Transactions, 9, 1966, pp 229-441.

3. Harris, T. A., "An Analytical Method to Predict Skidding in Thrust Loaded Angular Contact Ball Bearings", Journal of Lubrication Technology, Trans. ASME, Series F, Vol. 93, No. 1, 1971, pp 17-24.

4. Timoshenko, Strength of Materials Part II Advanced Theory and Problems, 3rd Edition, D. VanNostrand Co., Inc., 1958.

5. Crecelius, W. J. and Milke, D., "Dynamic and Thermal Analysis of High Speed Tapered Roller Bearings Under Combined Loading", Technical Report NASA CR 121207.

6. Harris, T. A., "How to Predict Bearing Temperature Increases in Rolling Bearings," Product Engineering, pp. 89-98, 9th December 1963.

7. Fernlund, I. and Andreason, S., "Bearing Temperatures Calculated by Computer", The Ball Bearing Journal No. 156, March 1969.

8. Andreason, S., "Computer Calculation of Transient Temperatures", The Ball Bearing Journal, No. 160, March 1970.

9. McCool, J. I., et al, "Technical Report AFAPL-TR-75-25, "Influence of Elastohydrodynamic Lubrication on the Life and Operation of Turbine Engine Ball Bearings - Bearing Design Manual", S K F Report No. AL75P014 submitted to AFAPL and NAPTC under AF Contract No. F33615-72-C-1467, Navy MIPR No. M62376-3-000007, May 1975.

10. Cheng, H. S., "Calculation of EHD Film Thickness in High Speed Rolling and Sliding Contacts", MTI Report 67-TR-24 (1967).

11. Chiu, Y. P., "An Analysis and Prediction of Lubricant Film Starvation in Rolling Contact Systems", ASLE Transactions, 17, pp 23-35 (1974).

12. Tallian, T. E., "The Theory of Partial Elastohydrodynamic Contacts", Wear, 21, pp 49-101 (1972).

13. McGrew, J. M., et al, "Elastohydrodynamic Lubrication Preliminary Design Manual", Technical Report AFAPL-TR-70-27, pp. 20-21, November 1970.

14. Fresco, G. P., et al, "Measurement and Prediction of Viscosity-Pressure Characteristics of Liquids", A Thesis in Chemical Engineering Report No. PRL-3-66, Department of Chemical Engineering College of Engineering, The Pennsylvania State University, University Park Pennsylvania.

15. Crecelius, W. J., Heller, S., and Chiu, Y. P., "Improved Flexible Shaft-Bearing Thermal Analysis with NASA Friction Models and Cage Effects", S K F Report No. AL76P003, February 1976.

16. Jakob, M. and Hawkins, G. A., "Elements of Heat Transfer", 3rd Ed., John Wiley & Sons, Inc., 1957.

17. Kent's Mechanical Engineering Handbook-Power Volume, John Wiley & Sons, Inc., 12th Ed., 1960, Chapter 3, p. 20.

18. Burton, R. A., and Staph, H. E., "Thermally Activated Seizure of Angular Contact Bearings", ASLE Trans. 10, pp. 408-417 (1967).

19. Liu, J. Y., Tallian, T. E., McCool, J. I., "Dependence of Bearing Fatigue Life on Film Thickness to Surface Roughness Ratio", ASLE Preprint 74AM-7B-1 (1974).

20. Archard, J. and Cowking, E., "Elastohydrodynamic Lubrication at Point Contact", Proc. Inst. Mech. Eng., Vol. 180, Part 3B, 1965-1966, pp. 47-56.

21. Dowson, D. and Higginson, G., "Theory of Roller Bearing Lubrication and Deformation", Proc. Inst. Mech. Eng., London, Vol. 177, 1963, pp. 58-69.

22. Loewenthal, S. H., Parker, R. J., and Zaretsky, E. V., "Correlation of Elastohydrodynamic Film Thickness Measurements for Fluorocarbon, Type II Ester and Polyphenyl Ether Lubricants", NASA Technical Note D-7825, National Aeronautics and Space Administration, Washington, D. C., November 19, 1974.

23. McCool, J. I., et al, "Interim Technical Report on Influence of Elastohydrodynamic Lubrication on the Life and Operation of Turbine Engine Ball Bearings", S K F Report No. AL73P014, submitted to AFAPL and NAPTC under AF Contract No. F33615-72-C-1467, Navy MIPR No. M62376-3-000007, July 1974.

24. Allen, C. W., Townsend, D. P., and Zaretsky, E. V., "New Generalized Rheological Model for Lubrication of a Ball Spinning in a Nonconforming Groove", NASA Technical Note D-7280, National Aeronautics and Space Administration, Washington, D. C., May 1973.

APPENDIX - A

S K F COMPUTER PROGRAM AT77Y001 "SHABERTH"

HIERARCHIAL FLOW CHART

APPENDIX A

S K F COMPUTER PROGRAM AT77Y001 FLOW CHART

Flow Chart

The hierarchical flow chart presents the program structure, listing the program elements in the order in which they would be called to solve the shaft-bearing dynamic, as well as steady state and transient temperature distribution problems. The various solution loops are indicated, as well as notes which indicate the functions of various subroutine groupings.

Each line in the flow chart represents a program element, subroutine, function or the main program ALWAYS. The call of one subroutine by another is denoted by indenting the called subroutine relative to the routine doing the calling. As an example, subroutine SKF calls subroutines FLAGS, TYPE, PROPST, LUPROP, LUBCON, DATOT, CNVRT, CONS and SPRING. Subroutine CONS calls CONBRI, BCON, TCON, CRCON and CONBRZ. BCON calls ABDEL. Both TCON and CRCON call ABDEL and SLICES.

The first mention of a subroutine within the flow chart includes the entire list of subordinate program elements. At subsequent calls to that subroutine the list of subordinate elements is omitted. As an example the first call to subroutine AXLBOJ is followed by the subordinate elements, JMVIKT, SNITMT, NUMLOS, DUBSIM, MEIE, MEIL and SIMQ. After the call of AXLBOJ from INDEL, the subordinate elements are not listed but are, nevertheless, employed. The list of subordinate program elements are omitted in repeated calls of subordinate GUESS, BEAR, SOLVXX and DELIV3 as well as AXLBOJ.

As noted earlier, rolling equilibrium is calculated, first without, then if required, with friction forces included. Whether or not friction is considered is highlighted with the words Frictionless or Friction.

If the Program is too large to fit in its entirety on the user's computer, segments of the program may be "overlaid". For this purpose the Program is subdivided into ten (10) moduleswhich can be sequentially "overlaid".

The Program segments SKF, TEMPIN, SHAFT and GUESS all perform initiation functions and with the exception of GUESS, are called only once per program execution.

The real problem solving portion of the program is embodied in segment ALLT. Within this segment the shaft bearing solution is obtained through the call to SHABE, then the steady state or transient temperature distributions are obtained.

This scheme is repeated until the end objective, steady state thermal equilibrium or time up for the transient scheme, is realized.

The nonlinear equation solver SOLV13 is central to the program and deserves special discussion as related to the flow chart. The first call to SOLV13 is from BEAR. Only for this first call are all of the SOLV13 subordinate subroutines listed as noted earlier. These include INSOLV, EQS. PARDER, SIMQ, EQCHEK, ERWRIT and ERCHEK. In the subsequent call to SOLV13 in which the steady state temperatures are being calculated, the above listed subroutines are again called but these calls with the exception of EQS are not listed on the flow chart.

EQS is the name given by SOLV13 to a subroutine which sets up the system of equations to be solved. EQS is brought into SOLV13 through the argument list. When the bearing equations are being solved, subroutine BRGGEQ is brought into SOLV13 and within SOLV13 is referenced by the name EQS. When the heat transfer equations are being solved as a consequence of the call of SOLV13 from ALLT, NET is brought into SOLV13 and is referenced as EQS.

S H A B E R T H FLOW CHART

```
ALWAYS
  SKF                                              Read and set bearing and bearing solution control data
    FLAGS
    TYPE
  ---------------------------------------------------------------------------------------------
    PROPST                                         Set material and lubricant properties and calculate
    LUPROP                                         constants for temperature dependency calculations
    LUBCON
  ---------------------------------------------------------------------------------------------
    DATOT                                          Write bearing input and hardcoded, preset data
      TITLE
      FITDAT
      ROLDAT
  ---------------------------------------------------------------------------------------------
    CNVRT                                          Calculate bearing related constants
    CONS
      CONBR1
      BCON
        ABDEL
      TCON
        ABDEL
        SLICES
      CRCON
        ABDEL
        SLICES
      CONBR2
    SPRING
  ---------------------------------------------------------------------------------------------
  TEMPIN                                           Read and write thermal and thermal solution control
    INDUM                                          data and calculate heat transfer coefficients
    RWHTC
    RWS
    RWHC
    TMAP
  ---------------------------------------------------------------------------------------------
  SHAFT                                            Read and write shaft geometry, loading and bearing
    ZERO                                           position data
    ARRANG
      ORDERR
    OUTINP
  ---------------------------------------------------------------------------------------------
    AXLBOJ                                         Calculate shaft deflection constants
      UNVIKT
      SNITMT                                       Make initial guesses of bearing displacements
      NUMLOS
        DURSIM
          MEIE
          MEIL
      SIMQ
    REACT
    INDEL
      AXLBOJ
  ---------------------------------------------------------------------------------------------
    PART                                           Calculate shaft influence coefficients
      PAR
        AXLBOJ
  ---------------------------------------------------------------------------------------------
  GUESS                                            Guess values of rolling element and cage variables,
    GRBRG                                          positions and speeds
      GROLL
    GBBRG
      GBALL
    STBRG
      GSTPOS
      GSTSPD
    GUESCG
    VARROC
  ---------------------------------------------------------------------------------------------
  ALLT                                             Begin the solution of the steady state or transient
                                                   thermal and temperature dependent shaft-bearing analyses
  ---------------------------------------------------------------------------------------------
    SHABE                                          Calculate Shaft-Bearing Equilibrium, in steps, for the
                                                   desired NPASS solution level
      FIT                                          Calculate bearing diametral clearance
        INTFIT
          SIMEQ
  ---------------------------------------------------------------------------------------------
      SONR1         PERFORM THESE SONR1 CALCULATIONS   Establish iteration scheme to satisfy shaft and bearing
        BEARC       FOR ALL NPASS VALUES.              inner ring equilibrium
  ---------------------------------------------------------------------------------------------
          BEAR                                     Establish iteration scheme to satisfy rolling element
            PREPAR                                 equilibrium.  Solve rolling element equilibrium,
        #4    INITX                                bearing by bearing, one element at a time
                UNLOAD
              MAXMIN
            SOLV13
              INSOLV
              EQCHEK
                DAMPCO
                ERCHEK
#1 #2 #3      EQS = BRGGEQ  ( WITHOUT FRICTION )   Evaluate ball-raceway contact loads, ball centrifugal
                EQBALL  ( BALL BEARING )           force and ball equilibrium equations
                  BGCTRL
                  BALLIN
                  BALLEQ
                EQTAPR  ( TAPERED ROLLER BEARING ) Evaluate tapered roller-raceway and flange contact
                  RGCTRL                           loads, centrifugal force and gyromoment and roller
                  TAPIN                            equilibrium equations
                  TAPEQ
                    TPNORM
                    FLNORM
                      PUSH
                EQCYL  ( CYLINDRICAL ROLLER BEARING )  Evaluate cylindrical roller-raceway and flange contact
                  RGCTRL                           loads, centrifugal force and roller equilibrium equations
                  ROLLIN
                    FLNDEF
                  ROLLEQ
                    TPNORM
                    FLNORM
                      PUSH
              PARDER                               Calculate partial derivatives of rolling element
                EQS = BRGGEQ                       equilibrium equations with respect to rolling element
              SIMQ                                 positions and calculate new positions
              EQCHEK
                EQS = BRGGEQ
                ERJRIT
            SUMF                                   Sum rolling element-inner ring forces and moments
              SUMK
              SUMFL
            LIFE
  ---------------------------------------------------------------------------------------------
                                                   Calculate bearing fatigue life
        BFILL                                      Add partial derivatives of bearing inner ring force
        SHAPA                                      with respect to inner ring displacement to the shaft
          FILLA                                    equilibrium equation and obtain new inner ring positions
        SIMEQ
  ---------------------------------------------------------------------------------------------
      BEAR      IF NPASS = 0, RETURN TO            After shaft equilibrium is satisfied, calculate bearing
      GUESS     ALLT AND PRINT OUTPUT.             reaction forces at the bearing equilibrium positions
  ---------------------------------------------------------------------------------------------
                                                   and estimate a new set of rolling element speeds
      VISCOS                                       Evaluate temperature dependent lubricant properties
        ALPHAO                                     and constants.
      DRAGNO
      STCON
  ---------------------------------------------------------------------------------------------
      EVALUT  ( IF NPASS = 1 )   SKIP IF NPASS = 2 OR 3   Evaluate bearing performance with estimated rolling
        PREPAR                                     element speeds.
        BRGGEQ  ( WITH FRICTION )
  ---------------------------------------------------------------------------------------------
          EQBALL                                   Evaluate ball-raceway contact loads.
            BGCTRL
            BALLIN
            BALLEQ
  ---------------------------------------------------------------------------------------------
              FMIX                                 Evaluate ball-raceway EHD film thickness, inlet and
                TINT                               concentrated contact friction forces and heat generation
                STAREC                             rate.
                THERFI
                HDFRIC
                  FBAR
```

#1 THERMAL EQUILIBRIUM OR TIME TRANSIENT CA

#2 BEARING CLEARANCE

#3 SHAFT BEARING EQUILIBRIUM

#4 HOLDING ELEMENT EQUILIBRIUM

```
              | ASLOAD
              | EHDSKF
              |   | FRICTN
              | FRICTN
          |--------------------------------------------
          | CAGESP                                      Evaluate cage speed, ball-cage normal and friction
          | CAGEEQ                                      forces and heat rates plus cage rail-ring land
          |   | CGLAND                                  normal and friction forces plus heat generation rates.
          |   |   | CGDRY
          |   |   | CGWET
          |   | CGRE
          |   |   | CGBAL1
          |   |   |   | CGNRMW
          |   |   |   |   | CGEHDP
          |   |   |   |   | CGHDP
          |   |   |   | CGFRW
          |   |   |   |   | CGHDF
          |   |   |   |   | CGEHI
          |   |   |   |   |   | EHDSKF
          |   |   |   |   |   |   | FRICTN
          |   |   |   | CGNRMD
          |   |   |   | CGFRD
          |   | SUMCGL
          |   | SUMRE
          |--------------------------------------------
          | BRGAX                                       Evaluate ball, position dependent, accelerations.
          |   | BRAX
          |--------------------------------------------
          | LIFE                                        Calculate bearing fatigue life
          |   | LRHS
          |--------------------------------------------
          | HUTRAC                                      Calculate ball-raceway heat transfer coefficients
          |   | HUT
        |----------------------------------------------
        | EQTAPR                                        Evaluate roller-raceway and roller-flange contact
          | BGCTRL                                      loads
          | TAPIN
          | TAPEQ
          |   | TPNORM
          |   | FLNORM
          |   |   | PUSH
          |--------------------------------------------
          |   | FMIXR                                   Evaluate roller raceway EHD film thickness inlet and
          |   |   | STARFC                              concentrated contact friction forces and heat generation
          |   |   | THERF1                              rate
          |   |   | HRFRIC
          |   |   | ASLOAD
          |   |   | EHDSKF
          |   |   |   | FRICTN
          |   |   | FRICTN
          |--------------------------------------------
          |   | FLMIX                                   Evaluate roller-flange EHD film thickness inlet and
          |   |   | STARFC                              concentrated contact friction forces and heat genera-
          |   |   | THERF1                              tion rate
          |   |   | HDFRIC
          |   |   | ASLOAD
          |   |   | EHDSKF
          |   |   |   | FRICTN
          |   |   | FRICTN
          |--------------------------------------------
          | CAGSPD                                      Evaluate tapered roller bearing cage speed, cage
          | CAGEEQ                                      forces, roller accelerations, bearing fatigue life,
          | BRGAX                                       roller-raceway and roller-flange heat transfer
          | LIFE                                        coefficients
          | HUTRAC
          | HUTFLN
        |----------------------------------------------
        | EQCYL                                         Evaluate cylindrical roller-raceway and flange normal
          | BGCTRL                                      and friction forces and heat generation rates plus
          | ROLLIN                                      EHD film thicknesses
          |   | FLNDEF
          | ROLLEQ
          |   | TPNORM
          |   | FLNORM
          |   |   | PUSH
          |   | FMIXR
          |   | FLMIX
          |--------------------------------------------
          | CAGSPD                                      Evaluate cylindrical roller bearing cage speed, cage
          | CAGEEQ                                      forces, roller acceleration, bearing fatigue life, plus
          | BRGAX                                       roller-raceway and roller-flange heat transfer
          | LIFE                                        coefficients
          | HUTRAC
          | HUTFLN      IF NPASS = 1, RETURN TO ALLT.
      |------------------------------------------------
      | BEAR  ( IF NPASS = 2 ) SKIP IF NPASS = 3        Solve the rolling element and cage equilibrium equations
      |   | PREPAR                                      with friction included, using the inner ring positions
      |#4| SOLV13                                       established by the frictionless solution
      |   |   | EQS = BRGGEQ  ( WITH FRICTION )
      |------------------------------------------------
      | SONR1  ( IF NPASS = 3 )                         Solve shaft, rolling element and cage equilibrium
      |   | BEARC                                       equations with friction included
      |#3|   | BEAR
      |   |   |   | PREPAR
      |   |   |#4| SOLV13
      |   |   |   |   | EQS = BRGGEQ  ( WITH FRICTION )
      |   |   | SUMF
      |   |   |   | SUMK
      |   |   |   | SUMFL
    |--------------------------------------------------
    | DELIV3                                            Write shaft-bearing output
    |   | TITLE
    |   | RITE
    |   | REOUT3
    |   |   | RITE2
    |   | REFLOT
    |   |   | RITE4
    |--------------------------------------------------
    | FILLGT .                                          Solve for the steady state temperature distribution.
    | SOLV13                                            Write shaft-bearing output and temperature map if
    |   | EQS = NET  ( FOR STEADY STATE THERMAL SOLUTION )   desired.
    |   |   | NETEET
    | DELIV3
    | TMAP
    |--------------------------------------------------
    | STEPMA                                            Determine an appropriate time step and solve for the
    |   | NET  ( FOR THE TRANSIENT THERMAL SOLUTION )   transient temperature distribution.  Write shaft-
    |   |   | NETEET                                    bearing output and temperature map if desired
    | DELIV3
    | NET
    |   | NETEET
    | TMAP
    |--------------------------------------------------
    | DELIV3                                            Write output for the final solution
    | TMAP
-------------------------------------------------------------
```

(Left vertical label: CALCULATIONS)

```
NPASS = 0     SHAFT AND BEARING INNER RING EQUILIBRIUM ARE SATIS-
              FIED CONSIDERING ELASTIC CONTACT FORCES.  NO LUBRI-
              CATION OR FRICTION EFFECTS ARE CONSIDERED.

NPASS = 1     SHAFT AND BEARING INNER RING EQUILIBRIUM ARE SATIS-
              FIED CONSIDERING ELASTIC CONTACT FORCES.  LUBRICATION
              AND FRICTION EFFECTS ARE CONSIDERED USING RACEWAY
              CONTROL ( BALL BEARINGS ) OR EPPICYCLIC ( ROLLER
              BEARINGS ) ASSUMPTIONS TO ESTIMATE ROLLING ELEMENT
              AND CAGE SPEEDS.

NPASS = 2     SHAFT AND BEARING INNER RING EQUILIBRIUM ARE SATIS-
              FIED CONSIDERING ELASTIC CONTACT FORCES.  USING THE
              INNER RING POSITIONS THUS OBTAINED, ROLLING ELEMENT
              AND CAGE EQUILIBRIUM ARE DETERMINED CONSIDERING
              FRICTION.

NPASS = 3     COMPLETE SOLUTION.  SHAFT AND BEARING INNER RING
              PLUS ROLLING ELEMENT AND CAGE EQUILIBRIUM ARE DETER-
              MINED CONSIDERING ALL ELASTIC AND FRICTION FORCES.
```

APPENDIX - B

HEAT TRANSFER COMPUTATION NOTES

APPENDIX B

HEAT TRANSFER INFORMATION

B.1 BACKGROUND

The temperature portion of Program AT77Y001 is designed to produce temperature maps for an axisymmetric mechanical system of any geometrical shape. The mechanical system is first approximated by an equivalent system comprising a number of elements of simple geometries. Each element is then represented by a node point having either a known or an unknown temperature. The environment surrounding the system is also represented by one or more nodes. With the node points properly selected, the heat balance equations can be set up accordingly for the nodes of unknown temperature. These equations become non-linear when there is convection and/or radiation between two or more of the node points considered. The problem is therefore reduced to solving a set of linear and/or non-linear equations for the same number of unknown nodal temperatures. It is obvious that the success of the approach depends largely on the physical subdivision of the system. If the subdivision is too fine, there will be a large number of equations to be solved; on the other hand, if the subdivision is too crude, the results may not be reliable.

In a system consisting of rolling bearings, for the sake of simplicity, the elements considered are usually axially symmetrical, e.g., each of the bearing rings can be taken as an element of uniform temperature. For an element which is not axially symmetrical, its temperature is also assumed to be uniform and its presence is assumed not to distort the uniformity in temperature of a neighboring element which is axially symmetrical. That is, the non-symmetrical element is represented by an equivalent axially symmetrical element with approximately the same surface area and material volume. This kind of approximation may seem to be somewhat unrealistic, but with properly devised equivalent systems, it can be used to solve complicated problems with results satisfying some of the important engineering requirements.

The computer program can solve the heat-balance equations for either the steady state or the transient state conditions and produce temperature maps for the mechanical system when the input data are properly prepared.

B.2 BASIC EQUATIONS

B.2.1 Heat Conduction

The rate of heat flow $q_{ci,j}$ (W) that is conducted from node i to node j may be expressed by,

$$q_{ci,j} = \frac{\lambda_{ij} A_{ij}}{L_{ij}} \quad (t_i - t_j)$$

t_i and t_j are the temperatures at i and j, respectively, $A_{i,j}$ the area normal to the heat flow, (m^2) L_{ij} the distance (m) and λ_{ij} the thermal conductivity between i and j, (W/m°C).

Assuming that the structure between point i and j is composed of different materials, an equivalent heat conductivity may be calculated as follows:

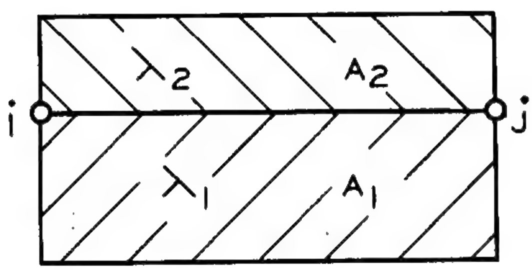


Fig. B-1

$$\lambda_{ij} = \frac{\lambda_1 A_1 + \lambda_2 A_2}{A_{ij}}$$

$$A_{ij} = A_1 + A_2$$

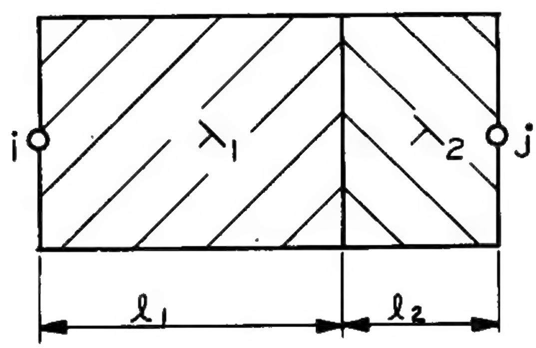


Fig. B-2

$$\lambda_{ij} = \frac{\ell_{ij}}{\ell_1/\lambda_1 + \ell_2/\lambda_2}$$

$$\ell_{ij} = \ell_1 + \ell_2$$

The calculation of the areas will be discussed in Section B.2.5.

B.2.2 CONVECTION

The rate of heat flow that is transferred between a solid structure and air by <u>free convection</u> may be expressed by

$$q_{vi,j} = \alpha_{i,j} A_{i,j} |t_i - t_j|^{1.25} \cdot \text{SIGN}(t_i - t_j)$$

where

$$\text{SIGN} = \begin{cases} 1, & \text{if } (t_i - t_j) \geq 0 \\ -1, & \text{if } (t_j - t_i) < 0 \end{cases}$$

in which

$$\alpha_{ij} = \begin{cases} 2.5 \cdot 10^{-2}\ \text{W/m}^2\text{-}(\text{degC})^{1.25} & \text{for hot surfaces facing upward and cold surfaces facing downward} \\ 1.4 \cdot 10^{-2}\ \text{W/m}^2\text{-}(\text{degC})^{1.25} & \text{for hot surfaces facing downward and cold surfaces facing upward} \\ 1.8 \cdot 10^{-2}\ \text{W/m}^2\text{-}(\text{degC})^{1.25} & \text{for vertical surfaces} \end{cases}$$

For other special conditions, α_{ij} must be estimated by referring to heat transfer literature.

The rate of heat flow that is transferred between a solid structure and a fluid by forced convection may be expressed by

$$q_{ni,j} = \alpha_{i,j} A_{i,j} (t_i - t_j)$$

in which α_{ij} is the heat transfer coefficient.

Now, with $\alpha = \alpha_{ij}$, introduce the Nusselt number

$$N_u = \frac{\alpha L}{\lambda}$$

the Reynolds number

$$R_e = \frac{UL}{\nu}$$

and the Prandtl number

$$P_r = \frac{\rho \nu C_p}{\lambda}$$

where

L is a characteristic length which is equal to the diameter in the case of a cylindrical surface and is equal to the plate length in case of a flat surface (m)

U is a characteristic velocity which is equal to the difference between the fluid velocity at some distance from the surface and the surface velocity (m/sec)

λ is the fluid thermal conductivity (W/M^o C)

ν is the fluid kinematic viscosity (M^2/sec)

ρ is the fluid density (kg/m^3)

c_p is the fluid specific heat (J/kg^{o}C)

For given values of R_e and P_r the Nusselt number N_u and thus the heat transfer coefficient may be estimated from one of the following expressions:

Laminar flow along a flat plate: $R_e < 2300$

$$N_u = 0.323 \sqrt{R_e} \cdot \sqrt[3]{P_r}$$

Laminar flow of a liquid in a pipe:

$$N_u = 1.36 \sqrt[3]{R_e \cdot P_r (\frac{D}{L})}$$

where D is the pipe diameter and L the pipe length

Turbulent flow of a liquid in a pipe:

$$N_u = 0.027 \cdot R_e^{0.8} \cdot \sqrt[3]{P_r}$$

Gas flow inside and outside a tube:

$$N_u = 0.3 R_e^{0.57}$$

Liquid flow outside a tube:

$$N_u = 0.6 R_e^{0.5} \cdot P_r^{0.31}$$

Forced free convection from the outer surface of a rotating shaft :

$$N_u = 0.11 \left[0.5 R_e^2 \cdot P_r \right]^{0.35}$$

where the Reynolds number R_e is developed by the shaft rotation.

$$R_e = \frac{\omega\pi D^2}{\nu}$$

in which ω is the angular velocity (rad/sec)
D is the roller diameter (m)

The average coefficient of forced convection to the lubricating oil within a rolling contact bearing may be approximated by,

$$\alpha = 0.0986 \left\{ \frac{N}{\nu} \left[1 \pm \frac{D \cos (\beta)}{d_m} \right] \right\}^{\frac{1}{2}} \lambda (P_r)^{1/3}$$

using + for outer ring rotation
- for inner ring rotation

in which N is the bearing operating speed (rpm)
D is the diameter of the rolling elements (mm)
d_m is the bearing pitch diameter (mm)
β is the bearing contact angle (degrees)

B.2.3 FLUID FLOW

The rate of heat flow that is transferred from fluid node i to fluid node j by fluid flow is

$$q_{fi,j} = \rho \dot{V}_{ij} \, C_p \, (t_i - t_j)$$

$\dot{V}_{ij}$ is the volume rate of flow from i to j. It must be observed that the continuity of mass requires the following equation to be satisfied

$$\Sigma \, \dot{V}_{ij} = 0$$

provided the fluid density is constant. The summation should be extended over all nodes i within the fluid which have heat exchange with node j by fluid flow.

B.2.4 HEAT RADIATION

The rate of heat flow that is radiated to node j from node i is expressed by

$$q_{Ri,j} = \delta_{i,j} \, \{ (t_i+273)^4 - (t_j+273)^4 \}$$

where

$$T_j = t_j + 273.16$$

$$T_i = t_i + 273.16$$

and the value of the coefficient $\delta_{i,j}$ depends on the geometry and the emissivity or the absorptivity of the bodies involved.

For radiation between large, parallel and adjacent surfaces of equal area, $A_{i,j}$ and emissivity, $\varepsilon_{i,j}$, $\delta_{i,j}$ is obtained from the equation

$$\delta_{i,j} = \varepsilon_{i,j} \quad \sigma A_{i,j}$$

where σ, the Stefan-Boltzmann constant, is

$$\sigma = 5.76 \cdot 10^{-8} \ W/m^2 \ /(degK)^4$$

For radiation between concentric spheres and coaxial cylinders of equal emissivity, $\varepsilon_{i,j}$, $\delta_{i,j}$ is given by the equation

$$\delta_{ij} = \frac{\varepsilon_{i,j} \, \sigma A_{i,j}}{1 + (1 - \varepsilon_{i,j}) \dfrac{A_{i,j}}{A^*_{i,j}}}$$

where σ is as above $A_{i,j}$ is the area of the enclosed body and $A^*_{i,j}$ is the area of the surrounding body, i.e. $A_{i,j} < A^*_{i,j}$.

Expressions for $\delta_{i,j}$ that are valid for more complicated geometries or for different emissivities may be found in the heat transfer literature.

B.2.5 CALCULATION OF AREAS

In the case of heat transfer in the axial direction $A_{i,j}$ is given by the equation (Fig. B-3)

$$A_{i,j} = 2\pi r_m \cdot \Delta r$$

Referring to the input instructions, Section V, but recalling L must be input in mm not m.

$$L_1 = r_m = \frac{r_1 + r_2}{2}$$

$$L_2 = \Delta r = r_2 - r_1$$

In the case of heat transfer in the radial direction, $A_{i,j}$ is obtained from the expression

$$A_{i,j} = 2\pi\ r_m \cdot H;\ L_1 = r_m;\ L_2 = H$$

and similarly for the radiation term above

$$A^*_{i,j} = 2\pi\ r^*_m H$$

$$L_3 = r^*_m$$

$$L_2 = 2H$$

in which H is the length of the cylindrical surface; where heat is conducted between i and j, r_m is given by the same equation as above (Fig. B-4 (a)); where heat is convected between i and j, r_m is the radius of the cylindrical surface (Fig. B-4(b)); where heat is radiated between i and j, r_m is the radius of the enclosed cylindrical surface and r_m* the radius of the surrounding cylindrical surface (Fig. B-4(c))

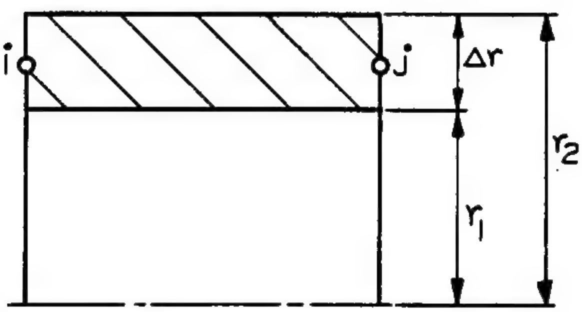


Fig. B-3

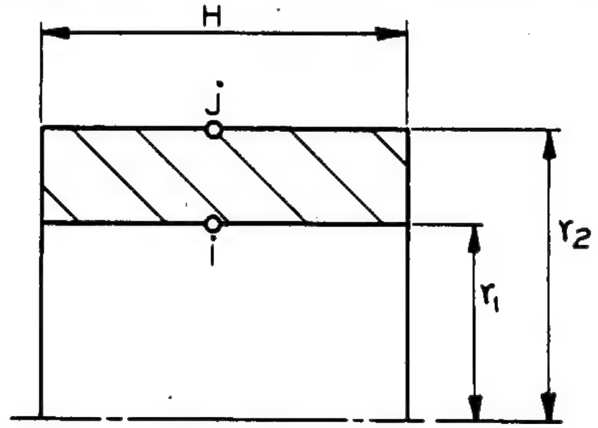


Fig. B-4(a)

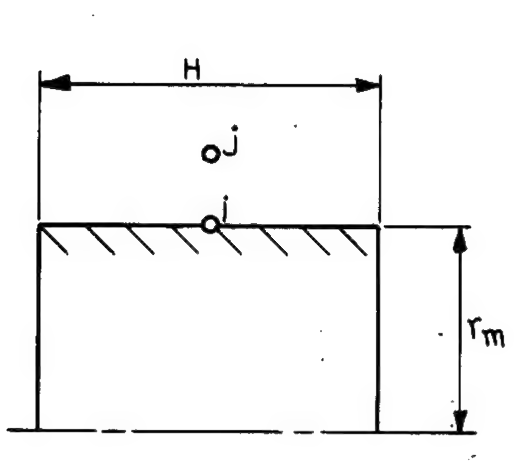


Fig. B -4(b)

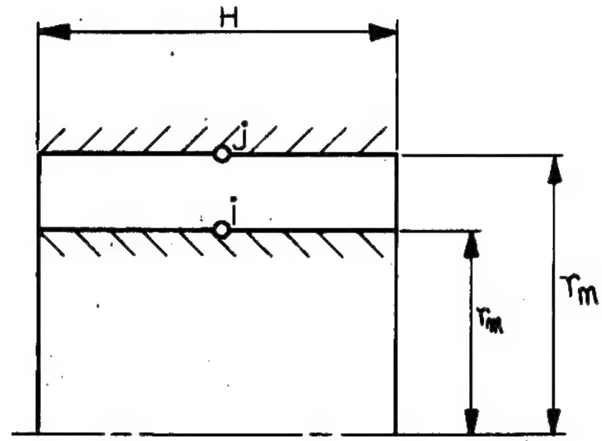


Fig. B -4(c)

B.3.1 TRANSIENT ANALYSIS

For the transient analysis all of the data pertaining to the node to node heat transfer coefficients must be provided by the input. Additionally, the volume and the specific heat at each node is required. For metal nodes this input is straightforward. However, when fluid flow is being considered there is no easy way to approximate the fluid nodal volume in a free space such as the bearing cavity. However, through use of the Program, the user's ability to make appropriate estimates will improve.

APPENDIX - C

S K F COMPUTER PROGRAM AT77Y001 "SHABERTH"

INPUT FORMAT FORMS

HOW TO READ AND USE THE FOLLOWING FORMS

PROGRAM NAMES OF INPUT ITEMS, FOR EXAMPLE:

																																		WI					
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
FORMAT, IF BLANK, WRITE																X	.	X	ANYWHERE IN THE FIELD, OR:																		X	X	X
YOU CAN WRITE YOUR DATA ON THIS LINE, FOR EXAMPLE:																																						3	6

					DIA									BD(5)							I2					A(6)							B(7,3)						
41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
OR:			X	.	X	X	E	+	X	TO THE RIGHT													X	X		MEANS INTEGER TO THE RIGHT.													
			5	.	6	3	E	+	2		3	.	6	1	1	E	+	1	1				1	6		2	0	.	1			1	1	.	3	6			

DESCRIPTION OF ITEM

DESCRIPTION OF ITEM

DESCRIPTION OF ITEM FOR EXAMPLE: BALL DIAMETER

EX. NO. OF BALLS

ALL VALUES ARE TO BE GIVEN IN THE UNITS OF THE SI-SYSTEM, THE INTERNATIONAL SYSTEM. EXCEPTIONS ARE LENGTHS WHICH ARE EXPRESSED IN MILLIMETERS INSTEAD OF METERS, AND PRESSURES WHICH ARE GIVEN IN NEWTONS/MM^2 INSTEAD OF NEWTONS/M^2. VOLUMES IN MM^3.

Title Card
Card Type TT1

Title(20) 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
2	0	A	4																																																																											

Title to be printed on each page.

TITLE CARD

CARD TYPE TT2 - One Card

Columns	Field	Format	Description
1–10	COV(1)	F10.0	SHAFT SPEED REVOLUTIONS PER MINUTE
11–15	IFLAG (1) NBRG	F5.0	NUMBER OF BEARINGS. E(4)
16–20	IFLAG (2) NPRINT	F5.0	PRINT FLAG IF NOT BLANK DEBUG OUTPUT WILL BE OBTAINED. IF = 1 MAIN LOOP DEBUG ONLY. IF = 2 FULL DEBUG OUTPUT. E(5)
21–25	IFLAG(3) ITFIT	F5.0	FIT CALC. FLAG, BLANK MEANS NO FIT CALCULATION AND NO ITERATION. POSITIVE NO. MEANS FIT ITERATION WITH MAX = 5 - NEGATIVE NO. MEANS FIT ITERATION MAX = GIVEN NO. E(6) Shaft Fit Call Flag 0 = None + = Smax -,= User
26–30	IFLAG(4) ITMAIN	F5.0	MAIN LOOP ITERATION FLAG. POSITIVE NO. MEANS PRE-SET MAX IS USED = 15 NEGATIVE NO. MEANS ITERATION LIMIT = GIVEN NO. BLANK = 15 ITERATION. E(7)
31–40	COV(2) EPSFIT	F10.0	FIT LOOP ACCURACY IT IS DEFINED AS CHANGE OR DIAMETRAL CLEARANCE DIVIDED BY ROLLER ELEMENT DIAMETER. IF LEFT BLANK, PRE-SET ACCURACY OF 0.0001 IS USED. E(1)
41–50	COV(3) EPS1	F10.0	MAIN LOOP ACCURACY FOR FRICTIONLESS SOLUTION. WHEN THE LATEST CHANGE OF BEARING DEFLECTION IS LESS THAN EPS1 OR EPS2 * BEARING DEFLECTION, OR THE BEARING ANGULAR DEFLECTION IS LESS THAN 2 x 10[illegible] AND LINEAR DEFLECTION IS LESS THAN 5×10^{-8}, THEN ITERATION IS STOPPED. IF LEFT BLANK, PRE-SET ACCURACY OF 0.001 IS USED. E(2)
51–60	COV(4) EPS2	F10.0	MAIN LOOP ACCURACY FOR END OR COULOMB SOLUTION. IF LEFT BLANK, PRE-SET ACCURACY OF 0.0001.
61–78			
79	IMT	X	Bearing Material Property Flag **
80	NPASS	X	Bearing Solution Level Flag*

*NPASS, IFLAG(5), Bearing Solution Level Flag

0 - Elastic Contact Forces are calculated, no lubrication or friction effects are considered

1 Elastic Contact Forces are calculated. Lubrication and friction effects are considered using raceway control (ball bearing) or epicyclic (roller bearing) assumptions to estimate rolling element and cage speeds

2 Inner Equilibrium is satisfied considering only the Elastic Contact Forces. Using the inner ring positions thus obtained, rolling element and cage equilibrium are determined considering friction

3 Complete Solution. The inner ring, rolling element and cage equilibrium is determined considering all elastic and friction forces

**IMT If one or more of the bearing ring or rolling element materials is other than steel, set IMT to 1 and include card types B11 through B14 for all bearings.

Bearing Data, One Set per Bearing. All applicable cards B1 through B16 for Bearing 1, followed by B1 through B16 for Bearing 2, etc.

CARD TYPE B1

Columns	Variable	Format	Description
1–10	BD(1)	A1	Bearing Type BALL CYLINDRICAL TAPERED Must Begin in Column 1
11–30		5A4	Ring Material Designation — Inner Ring
31–50		5A4	Ring Material Designation — Outer Ring
51–60	BD(43)	F10.0	Ring Life Factors — Outer
61–70	BD(44)	F10.0	Ring Life Factors — Inner
71–78	BD(2)	F8.0	Orientation Angle in Degrees of Rolling Elements 0.0 or Blank means first element is on the Y-axis.
80	NC		Crown Drop Flag

NC Normally zero or blank, If set to 1, non uniform roller-raceway profile geometry is input on card types B5 and B6.

BEARING DATA
CARD TYPE U2, One Card

Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
BD(5)	BD(6)	BD(7)	BD(8)	BD(29)	BD(3)		
Pitch Dia. (mm) Omit for Tapered Roller Bearing	No. of Rolling Elements	Diametral Clearance (mm)	Contact Angle (Deg)	Flange Angle (Tapered Only) See Fig.3.3 (Deg)	Outer Ring Speed (RPM)		
		Axial Play for Tapered. Single Bearing Only (mm)	½ Cup Incl. Angle for Tapered.				

BEARING DATA CARD TYPE B4 One Card-

BD (4)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80

BALL BEARING

Ball Dia. (mm)

103

BD (4) BD (10) BD (11) BD (12) BD (20) BD (21)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80

ROLLER BEARING

Roller Diameter (mm)	Roller Length End to End (mm)	Roller End Sphere Radius (mm)	Roller Included Angle (Deg)	Roller Crown Radius (mm)	Roller Flat Length (mm)
Diameter of Roller Large End for Tapered					

BEARING DATA
CARD TYPE B4A - ONE CARD

BD (4) BD (22)

1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80

BALL BEARING

Outer Raceway Curvature $f_o = Ro/D$	Inner Raceway Curvature $f_i = Ri/D$

BD (23) BD (24) BD (25) BD (26) BD (27) BD (28) BD (22)

1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80

ROLLER BEARING

Roller Raceway Effective Length Outer (mm)	Roller Raceway Effective Length Inner (mm)	Raceway Crown Radius Outer (mm)	Raceway Crown Radius Inner (mm)	Roller Corner or Raceway Undercut Relief Outer (mm)	Roller Corner or Raceway Undercut Relief Inner (mm)	No. of Slices

BEARING DATA
CARD TYPE B4B - Include Only for Cylindrical Bearings

BD (31)	BD (32)	BD (33)	BD(34)	BD (35)	
1 2 3 4 5 6 7 8 9 10	11 12 13 14 15 16 17 18 19 20	21 22 23 24 25 26 27 28 29 30	31 32 33 34 35 36 37 38 39 40	41 42 43 44 45 46 47 48 49 50	51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80
Flange Angle Outer Ring (Deg)	Flange Angle Inner Ring (Deg)	Outer Ring End Play (mm)	Inner Ring End Play (mm)	Flange Inversion Flag.	

Omit Cards 5 and 6 if crown drop flag is 0
Bearing Data
Card Type 5, as many as needed, maximum of 3, for the inner race.

RLAM(1,5)	(2,5)	(3,5)	(4,5)	(5,5)	(6,5)	(7,5)	
1 2 3 4 5 6 7 8 9 10	11 12 13 14 15 16 17 18 19 20	21 22 23 24 25 26 27 28 29 30	31 32 33 34 35 36 37 38 39 40	41 42 43 44 45 46 47 48 49 50	51 52 53 54 55 56 57 58 59 60	61 62 63 64 65 66 67 68 69 70	71 72 73 74 75 76 77 78 79 80
Crown Drop at Laminum 1 (mm)	Crown Drop at Laminum 2	... and so on, use more than one card if no. of laminae is greater than 7.					

Card Type 6, as many as needed maximum of 3, for the outer race.

RLAM(1,4)	(2,4)	(3,4)	(4,4)	(5,4)	(6,4)	(7,4)	
1 2 3 4 5 6 7 8 9 10	11 12 13 14 15 16 17 18 19 20	21 22 23 24 25 26 27 28 29 30	31 32 33 34 35 36 37 38 39 40	41 42 43 44 45 46 47 48 49 50	51 52 53 54 55 56 57 58 59 60	61 62 63 64 65 66 67 68 69 70	71 72 73 74 75 76 77 78 79 80
Crown Drop at Laminum 1 (mm)	Crown Drop at Laminum 2	...and so on					

Bearing Data

Card Type B7A, one card, Omit if NPASS, Title Card 2, is zero or blank.

BD(13)	BD(14)	BD(15)	BD(16)	BD(17)	BD(18)		
CLA Roughness Surface (microns)			Asperity Slope (Degrees)				
Outer	Inner	Rolling Element	Outer	Inner	Rolling Element		

BEARING DATA
CARD TYPE B7B - INCLUDE ONLY FOR ROLLER BEARINGS

BD(225)	BD(226)	BD (227)	BD (228)	BD (229)	BD (230)	
1 2 3 4 5 6 7 8 9 10	11 12 13 14 15 16 17 18 19 20	21 22 23 24 25 26 27 28 29 30	31 32 33 34 35 36 37 38 39 40	41 42 43 44 45 46 47 48 49 50	51 52 53 54 55 56 57 58 59 60	61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80
Surface Roughness Outer Ring Flange (Microns)	Surface Roughness Inner Ring Flange (Microns)	Surface Roughness Rolling Element End (Microns)	Asperity Slope Outer Ring Flange (Deg)	Asperity Slope Inner Ring Flange (Deg)	Asperity Slope Roller End	

Bearing Data

Card Type B8, one card, Omit if NPASS, Title Card 2, is zero or Blank

BD(36)	BD(37)	BD(38)	BD(39)	BD(40)	BD(41)		
1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Cage Type -1. for outer-ring land riding +1. for inner-ring land riding 0. for rolling element riding	Rail-Land Diameter (mm)	Single Rail Width (mm)	Rail-Land Diametral Clearance (mm)	Rolling Element-Cage Pocket Diametral Clearance (mm)	Cage Weight Kg.		

Bearing Data
Omit Cards B9 and B10 if operating clearances are not to be calculated
Card Type B9, one card

BD(51)	BD(52)	BD(53)	BD(54)	BD(55)	BD(56)		
Shaft Fit Positive if interference (mm)	Housing Fit, Positive if Interference (mm)	Shaft Effective Length (mm)	Bearing Inner Ring Width (mm)	Bearing Outer Ring Width (mm)	Housing Effective Width (mm)		

Card Type B10, one card

BD(57)	BD(58)	BD(60)	BD(61)	BD(62)	BD(64)		
Shaft Inner Diameter (mm)	Bearing Bore Diameter (mm)	Bearing Inner Ring Mean Outer Diameter (mm)	Bearing Outer Ring Mean Inner Diameter (mm)	Bearing Outer Diameter (mm)	Housing Outer Diameter (mm)		

Bearing Data
Omit Card Types B11 through B14 if Operating Clearances are not to be Calculated.

Card Type B11, one card

BD(66)	BD(67)	BD(68)	BD(69)	BD(70)			
Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
Modulus of Elasticity N/mm^2							
Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

Card Type B12, one card

BD(71)	BD(72)	BD(73)	BD(74)	BD(75)			
Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
Poisson's Ratio							
Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

Card Type B13, one card

BD(76)	BD(77)	BD(78)	BD(79)	BD(80)			
Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
Density gm/cm^3							
Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

Card Type B14, one card

BD(81)	BD(82)	BD(83)	BD(84)	BD(85)			
Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
Coefficient Thermal Expansion 1/°C							
Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

BEARING DATA
CARD TYPE B15, one card, Omit if NPASS, Title Card 2, is Zero or Blank

BD (102)	BD (103)	BD (104)	BD (106)	BD (107)	BD (108)	BD (109)	BD (110)
1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Lubricant Replenishment Layer Thickness (mm)		Percent Lubricant in Bearing Cavity	Raceway-Rolling Element Asperity Friction Coefficient	Cage Friction Coefficient	Outer Race Flange. Repl. Layer Thickness.	Inner Race Flange Repl. Layer Thickness.	Flange-Roller End Friction Coefficient
Outer Raceway	Inner Raceway						

Bearing Data

Card Type B1b, one card. Omit if NPASS, Title Card 2, is Zero or Blank

Lubricant Properties

Columns	Variable	Format	Description
1–2	BD(93)	F2.0	Lubricant N Code 1-4 right justified NCODE
3–20		x A2	Lubricant Designation
21–30	BD(94)		VIS1 Dynamic Viscosity (cs) @ 100°F
31–40	BD(95)		VIS2 Dynamic Viscosity (cs) @ 210°F
41–50	BD(96)		RHO60 Density gm/cm³ @ 15.5°C
51–60	BD(97)		G Coefficient of Thermal Expansion 1/°C
61–70	BD(98)		Thermal Conductivity (W/M°C)
71–80			

If NCODE is zero read this data

Thermal Data
Card Type T1
ONE CONTROLLING CARD. IF NO TEMP. CALCULATION IS DESIRED, THEN LEAVE THE CARD BLANK.

Field	Card columns	Format	Group	Description
N	1–5	I5	GENERAL	HIGHEST NODE NUMBER
[illegible]	6–10	I5	GENERAL	HIGHEST UNKNOWN NODE NUMBER, SAME AS NUMBER OF UNKNOWN NODES
TEMP	11–15	F5.0X	GENERAL	COMMON INITIAL TEMPERATURE (°C). SELECTED TEMPERATURES CAN BE GIVEN DIFFERENT INITIAL VALUES USING CARD T3.
IPUNCH	16–20	I5	GENERAL	PUNCH FLAG, USUALLY ZERO. IF ≠ 0 FINAL TEMPERATURES WILL BE PUNCHED ACCORDING TO THE FORMAT OF CARD T3. THEN, THEY CAN BE READ IN AS INITIAL TEMPERATURES IN A LATER RUN.
IOB	21–25	I5	GENERAL	OUTPUT FLAG, USUALLY ZERO. IF ≠ 0 BEARING OUTPUT AND A TEMPERATURE MAP WILL BE PRINTED EVERY TIME THE SHAFT-BEARING PROGRAM HAS BEEN CALLED.
IT1	26–30	I5	GENERAL	MAXIMUM NO. OF CALLS OF THE SHAFT-BEARING PROGRAM.
EP1	31–35	F5.0X	STEADY STATE ONLY	ABSOLUTE ACCURACY OF TEMPERATURES
IT2	36–40	I5	STEADY STATE ONLY	ITERATION LIMIT, USUALLY LEFT BLANK. IF ≠ 0 THE INTERMEDIATE ITERATION LIMIT. IF BLANK PRE-SET LIMIT IS USED
EP2	41–45	F5.0X	STEADY STATE ONLY	ACCURACY USUALLY LEFT BLANK. IF ≠ 0 IT IS THE INTERMEDIATE REL. ACCURACY
START	46–50	F5.0X	TRANSIENT ONLY	STARTING TIME
STOP	51–55	F5.0X	TRANSIENT ONLY	STOPPING TIME
STEP IN	56–60	F5.0X	TRANSIENT ONLY	CALCULATION TIME STEP. IF LEFT BLANK, THE PROGRAM WILL CALCULATE A SUITABLE STEP.
TTIME	61–65	F5.0X	TRANSIENT ONLY	TIME INTERVAL BETWEEN PRINTED TEMP. MAPS. THE INTERVAL WILL ALWAYS BE AT LEAST EQUAL TO THE CALC. TIME STEP.
STIME	66–70	F5.0X	TRANSIENT ONLY	TIME INTERVAL BETWEEN CALLS OF SHAFT-BEARING PROGRAM. ALWAYS AT LEAST EQUAL TO THE CALC. TIME STEP.
	71–80			

Thermal Data - Bearing Temperatures

T2. One Card/Brg. Use only if no temperature calc. is desired, and then give no more thermal data.

1 2 3 4 5	6 7 8 9 10	11 12 13 14 15	16 17 18 19 20	21 22 23 24 25	26 27 28 29 30	31 32 33 34 35	36 37 38 39 40	41 42 43 44 45	46 47 48 49 50	51 52 53 54 55	56 57 58 59 60	61 62 63 64 65	66 67 68 69 70	71 72 73 74 75	76 77 78 79 80
Outer Race	Inner Race	Bulk Oil	*Flange #1	*Flange #2	*Flange #3	*Flange #4	Cage	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing			

* See Fig. 3.5 for the Flange numbering scheme. Note that a tapered roller bearing Flange is Flange #1.

Ball Bearings are not considered to have Flanges.

Thermal Data - Individual Initial Temperatures

T3. As many cards as needed, followed by a blank card

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
				x										x																																																													x					

Node Number	Initial Temperature	Etc. --	Node Number	Initial Temperature

Thermal Data - Bearing Node Numbers

T4, One card/brg.

1 2 3 4 5	6 7 8 9 10	11 12 13 14 15	16 17 18 19 20	21 22 23 24 25	26 27 28 29 30	31 32 33 34 35	36 37 38 39 40	41 42 43 44 45	46 47 48 49 50	51 52 53 54 55	56 57 58 59 60	61 62 63 64 65	66 67 68 69 70	71 72 73 74 75 76 77 78 79 80
x	x	x	x	x	x	x	x	x						
Outer Race	Inner Race	Bulk Oil	Flange #1	Flange #2	Flange #3	Flange #4	Cage	Shaft	Inner Ring	Rolling Element	Outer Ring	Housing		

See Fig. 3.5 for the Flange numbering scheme. Note that a tapered roller bearing inner ring Flange is Flange #3.

Ball Bearings are not considered to have Flanges.

Thermal Data - Nodes Where Bearing Frictional Heat is Generated

T5, One Card/brg.

Outer Race		Inner Race		Drag		Cage-Roll Element		Cage - Land							
Half the heat is generated at this node.	Half the heat is generated at this node.														
The above two node numbers can be the same number															

T5A, One Card/Brg if required.

Flange #1		Flange #2		Flange #3		Flange #4									

See Fig. 3.5 for the flange numbering scheme. Note that a tapered roller bearing inner ring flange is Flange #1.

Ball Bearings are not considered to have flanges.

Thermal Data - Nodes with a constant power source

'T6, as many cards as needed, followed by a blank card

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
			x	x									x	x									x	x									x	x									x	x									x	x									x	x									x	x					

Node Number	Power														

DATA CARD T7

HEAT TRANSFER COEFFICIENTS

ONE OR TWO CARDS/COEFFICIENT, AS MANY AS NEEDED, FOLLOWED BY A BLANK CARD.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
								I	I																																																																								

FIELD FOR LATER IDENTIFICATION OF THE HEAT TR. COEFFICIENT 1-10 $q = \lambda A/\ell \Delta t$ 11-20 $q = \alpha A \Delta t^{n}$ 21-30 $q = \alpha A \Delta t$ 31-40 $q = \varepsilon A (T_L^4 - T_H^4)$ 41-50 FLUID FLOW	 (CONDUCTIVITY W/M°C) (FREE CONV. NO.) α (FORCED CONV. NO.) (EMISSIVITY COEFF.) ε $\rho \cdot C_p \cdot \dot{V}$ (W/°C)	 (1 IF BLANK = 1,25) (ρ IS DENSITY, KG/M^3, CP SPECIFIC HEAT, WSEC/KG°C, $\dot{V}$ VOLUME FLOW M^3/SEC)					

THE FORCED CONVECTION NO. α CAN BE CALCULATED BY THE PROGRAM FROM THE FORMULA $\alpha = \lambda_{...}/L\ Nu$, WHERE $Nu = K \cdot R_e^A \cdot P_R^B$, $R_e = (U \cdot L \cdot \rho)/\eta$, $P_R = (\eta\ cp)/\lambda_{...}$, η = CONSTANT, OR $\eta = \eta(TEMP)$, η = DYNAMIC VISCOSITY. THEN THE FOLLOWING DATA MUST BE GIVEN AND A SECOND CARD MUST IMMEDIATELY FOLLOW. USE ONE OF THE 3 OPTIONS.

21-30 (α = CONST.) 21-30 (α = f(...), $\eta = \eta(t)$) 21-30 (α = f($\eta(t)$))	K K K	A A A	B BLANK B	L METER BLANK L METER	U(M/SEC) BLANK U	λ (W/M-°C) BLANK λ	(OPTION 1) (OPTION 2) (OPTION 3)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80

DYNAMIC VISCOSITY (N/M^2.S) BLANK BLANK	DENSITY (KG/M^3) BLANK DENSITY	SPECIFIC HEAT (WS/KG°C) BLANK SPECIFIC HEAT	BLANK LOW TEMP. T_L °C T_L	BLANK DYNAMIC VISCOSITY AT T_L DYNAMIC VISCOSITY AT T_L	BLANK HIGH TEMP. T_H (°C) T_H	BLANK DYNAMIC VISCOSITY AT T_H DYNAMIC VISCOSITY AT T_H	(OPTION 1) (OPTION 2) (OPTION 3)

THERMAL DATA - HEAT FLOW PATHS
I5, AS MANY CARDS AS NEEDED, MAX. 500, FOLLOWED BY A BLANK CARD
INDEX

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
								X	X				X	X				X	X																																																												

INDEX (INDAB = |INDEX|)	NODE I	NODE J				
1 ≤ INDAB ≤ 10	NODE I	NODE J	L_1 mm	L_2 mm	L_3 mm	CONDUCTION BETWEEN I AND J. AREA = $2\pi L_1L_2$. IF INDEX<0 AREA = L_1L_2, DISTANCE I-J = L_3.
11 ≤ INDAB ≤ 20	NODE I	NODE J	L_1	L_2	BLANK	NATURAL CONVECTION BETWEEN I AND J. AREA = $2\pi L_1L_2$. IF INDEX<0 AREA = L_1L_2.
21 ≤ INDAB ≤ 30	NODE I	NODE J	L_1	L_2	BLANK	FORCED CONVECTION BETWEEN I AND J. AREA AS ABOVE. IF $\eta(\tau)$, τ IS τ J.
31 ≤ INDAB ≤ 40	NODE I	NODE J	L_1	L_2	(L_3)	RADIATION BETWEEN I AND J. AREA AS ABOVE. FOR DESCRIPTION OF L_3, SEE USER'S MANUAL.
41 ≤ INDEX ≤ 50	NODE I	NODE J	INDEX OF FLUID FLOW AT NODE J, 41 ≤ INDEX ≤ 50	BLANK	BLANK	FLUID FLOW FROM NODE I TO NODE J. FIRST INDEX IS INDEX OF FLUID FLOW AT NODE I. Second index represents fluid flow going from node I to node J.
INDEX = 51	NODE I	NODE J	BEARING NUMBER BN 1 ≤ BN ≤ 5	RACEWAY FLAG 1, INNER RACE CONTACT 2, OUTER RACE CONTACT 3, FLANGE CONTACT#1 4, FLANGE CONTACT#2 5, FLANGE CONTACT#3 6, FLANGE CONTACT#4	FACTOR, USUALLY = 1. IF I OR J IS A NODE IN THE OIL BETWEEN THE CONTACTING SURFACES, THE FACTOR IS 0.5	CONDUCTION THROUGH A BEARING

Thermal Data - Node heat capacities, only for transient calculations

T9, One Card/Node

1-10	11-20	21-30	31-40	41-50	51-60	61-70	71-80
x x (cols 9, 10)							
Node Number	Volume at Node = $L_1 \cdot L_2 \cdot L_3$			Density (KG/M^3)	Specific Heat WS/KG°C		
If it is given without a sign, rotational symmetry is assumed and the volume is multiplied by 2π. If it is negative the volume is not changed.	L_1 mm	L_2 mm	L_3 mm				

Shaft Data - Shaft Geometry
Omit all shaft data cards if there are no bearings

51. as many cards as needed.

Columns 1–5	Columns 6–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
1								
Type of Shaft card. This card type is one.	Blank	X-coordinate of shaft section (mm)	Shaft Inner diameter immediately before the section. (mm)	Shaft Inner diameter immediately after the section. (mm)	Shaft Outer diameter immediately before the section. (mm)	Shaft Outer diameter immediately after the section. (mm)	Shaft Modulus of elasticity. If left blank, steel is assumed. 2.041×10^5 N/mm^2	

SHAFT DATA - BEARING POSITION AND SUPPORT, X-Y PLANE

S2, AS MANY CARDS AS THE NUMBER OF BEARINGS. ARRANGED IN ORDER OF INCREASING X-COORDINATE.

Card columns	Entry	Description
1–5	2 (col. 5)	TYPE OF SHAFT CARD. THIS CARD TYPE IS 2.
6–10		BLANK
11–20		X-COORDINATE OF BEARING POSITION. THE POSITION IS DEFINED AS THE LOCATION OF POINT A, WHICH IS FOR BALL BEARINGS THE INNER RING GROOVE CURVATURE CENTER, AND FOR ROLLER BEARINGS THE MIDDLE OF THE INNER RING CONTACT AREA. (mm)
21–30		RADIAL MOUNTING ERROR OF THE OUTER RING IN THE Y-DIRECTION. (mm)
31–40		RADIAL SPRING CHARACTERISTICS OF THE HOUSING IN THE Y-DIRECTION, MM/N.
41–50		ANGULAR MOUNTING ERROR OF THE OUTER RING IN THE Z-DIRECTION (radians)
51–60		ANGULAR SPRING CHARACTERISTICS OF THE HOUSING IN THE Z-DIRECTION, RADIANS/NMM.
61–80		

Shaft Data - Shaft Loading, X-Y plane forces, Z-moments

S3, as many cards as needed, followed by a blank card.

Columns 1–5	Columns 6–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
x (col. 5)								
3 (col. 5)								
Type of Shaft card. This card is type 3.	Blank	X-Coordinate of shaft section (mm)	Force Concentrated force Fy. N	Concentrated Moment Mz. N-mm	Distributed Load. Load intensity immediately before the section, N/mm	Distributed Load. Load intensity immediately after the section, N/mm	Concentrated axial load, N	Y-Coordinate for the line of action of the axial load. (mm)

Shaft Data - Bearing position and support, X-Z plane

S2, as many cards as needed. Card needed only if any data is different from X-Y plane

1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80

Type of shaft card This card type is 2.	Bearing number	Blank	Radial Mounting error of the outer ring in the Z-direction (mm)	Radial spring characteristics of the housing in the Z-direction mm/N.	Angular mounting error of the outer ring in the Y-direction. (radians)	Angular spring characteristics of the housing in the Y-direction radians/Nmm.		
	x							
2								

Shaft Data - Shaft Loading, X-Z plane forces, Y-moments.

S3, as many cards as needed, followed by a blank card.

Columns 1–5	Columns 6–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
x (col. 5)								
3 (col. 5)								
Type of shaft card. This card is type 3	Blank	X-coordinate of shaft section. (mm)	Concentrated force Fz. N	Concentrated moment My. N-mm	Distributed load. Load intensity immediately before the section, N/mm	Distributed load. Load intensity immediately after the section, N/mm	Concentrated axial load. N	Z-coordinate for the line of action of the axial load (mm)

APPENDIX - D

COMPUTER PROGRAM SHABERTH

SAMPLE OUTPUT

The sample SHABERTH output represents a steady state thermal analysis of a helicopter input pinion shaft which delivers turbine power to the transmission.

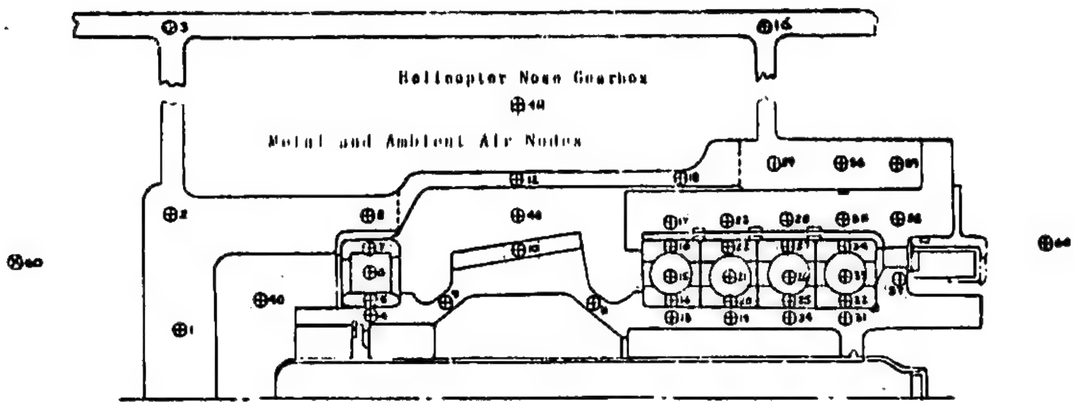
The system contains one 216 size cylindrical roller and a stack of four 6216 size split inner ring ball bearings. The bearing heat rates were calculated executing SHABERTH at the NPASS=1 level, with two clearance change iterations permitted for each each thermal iteration. The bearings act as heat sources which supply heat to the temperature nodes representing inner and outer rings, rolling elements and lubricant. Additional heat sources include a gear mesh and a seal which are specified by constant heat rate values at input. The nodal arrangement and the system equilibrium temperatures are shown in Fig. D-1.

All system temperatures were set at an estimated value of 75^{o}C. Bearing performance is determined based upon these temperatures. The bearing heat rates serve as input to the temperature calculation scheme which in turn produces a new set of system temperatures. These new temperatures affect bearing clearance and most notably lubricant viscosity in the next calculation of shaft bearing system performance parameters. Six iterations, calculations of bearing system performance, are required before the equilibrium condition is achieved. Equilibrium is satisfied when a check of all system temperatures, comparing two sequential iterations reflect less than a 1^{o}C change.

Although it is not included, shaft-bearing system output data may optionally be printed after each calculation of bearing frictional heat rates. This data allows the user to observe system performance as a function of temperature.

Since the program input is printed, the thermal data input serves as a good example, to be reviewed when preparing thermal data for another system.

Fig. D-1 Sample Problem Temperature Node Arrangement and Steady State Temperatures.

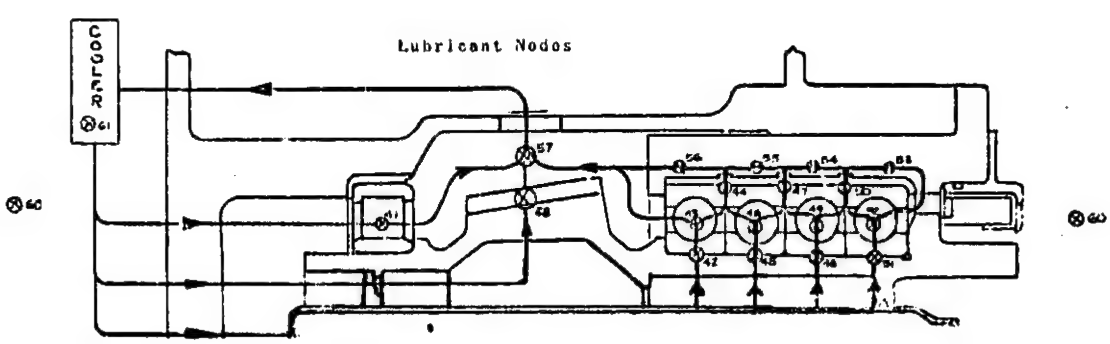


CALCULATED TEMPERATURES

ODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
1	93.103	2	94.718	3	80.748	4	115.494	5	117.314
6	113.694	7	106.140	8	102.353	9	107.245	10	106.375
11	95.107	12	101.027	13	78.735	14	82.319	15	86.455
16	84.951	17	86.153	18	88.457	19	79.555	20	84.628
21	94.332	22	88.084	23	87.388	24	80.050	25	85.253
26	94.409	27	88.393	28	87.313	29	86.495	30	79.623
31	81.989	32	87.267	33	95.328	34	88.343	35	86.504
36	85.144	37	103.561	38	84.396	39	83.695	40	92.442
41	101.703	42	72.266	43	78.789	44	90.902	45	72.494
46	84.750	47	94.703	48	72.632	49	85.078	50	95.285
51	73.183	52	86.022	53	95.677	54	95.415	55	95.130
56	93.922	57	101.778	58	87.757				

KNOWN BOUNDARY TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
59	75.000	60	24.000	61	70.000				



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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

THIS DATA SET CONTAINS 5 BEARINGS

BEARING NO. (1) - CYLINDRICAL ROLLER BEARING

BEARING NO. (2) - BALL BEARING

BEARING NO. (3) - BALL BEARING

BEARING NO. (4) - BALL BEARING

BEARING NO. (5) - BALL BEARING

THE MAXIMUM NUMBER OF FIT ITERATIONS ALLOWED IS 2 AND THE RELATIVE ACCURACY REQUIRED IS .00010

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CENTIGRADE, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CENTIGRADE.

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	16	.000	110.000	.038	.000	13600.	0.
2	15	.000	110.000	.022	23.000	13600.	0.
3	15	.000	110.000	.022	23.000	13600.	0.
4	15	.000	110.000	.022	23.000	13600.	0.
5	15	.000	110.000	.022	-23.000	13600.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	OUTER RING LAND RIDING	.952500	2.7500	120.3000	1.420	.200000
2	INNER RING LAND RIDING	.952500	2.8800	102.0000	.635	.180000
3	INNER RING LAND RIDING	.952500	2.8800	102.0000	.635	.180000
4	INNER RING LAND RIDING	.952500	2.8800	102.0000	.635	.180000
5	INNER RING LAND RIDING	.952500	2.8800	102.0000	.635	.180000

S T E E L D A T A

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	M-50	5.000	M-50	5.000
2	M-50	5.000	M-50	5.000
3	M-50	5.000	M-50	5.000
4	M-50	5.000	M-50	5.000
5	M-50	5.000	M-50	5.000

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ROLLING ELEMENT DATA

BEARING NUMBER (1) TYPE - CYLINDRICAL ROLLER BEARING

ROLLER DIAMETER	ROLLER LENGTH	ROLLER END SPHERE RADIUS	ROLLER INCL. ANGLE	AXIAL PLAY OUTER RING	AXIAL PLAY INNER RING	FLANGE ANGLE OUTER RING	FLANGE ANGLE INNER RING
17.0000	19.0000	1000.0000	.000	.0000	.0000	.000	.000

OUTER RACEWAY EFF.LENGTH	OUTER RACEWAY FLAT LENGTH	OUTER RACEWAY CROWN RAD.	INNER RACEWAY EFF.LENGTH	INNER RACEWAY FLAT LENGTH	INNER RACEWAY CROWN RAD.	NO. OF AXIAL LAMINAE
17.0000	8.7500	1524.000	17.0000	8.7500	1524.000	10

BEARING NUMBER (2) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
19.0500	.515	.520

BEARING NUMBER (3) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
19.0500	.515	.520

BEARING NUMBER (4) TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
19.0500	.519	.520

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R O L L I N G   E L E M E N T   D A T A

BEARING NUMBER (5)   TYPE - BALL BEARING

BALL DIAMETER	OUTER RACEWAY CURVATURE	INNER RACEWAY CURVATURE
19.0500	.515	.520

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

S U R F A C E D A T A

BEARING NUMBER	CLA ROUGHNESS OUTER	INNER	ROLL. ELM.	RMS ASPERITY SLOPE OUTER	INNER	ROLL. ELM.
1	.20	.20	.20	2.000	2.000	2.000
2	.15	.15	.15	2.000	2.000	2.000
3	.15	.15	.15	2.000	2.000	2.000
4	.15	.15	.15	2.000	2.000	2.000
5	.15	.15	.15	2.000	2.000	2.000

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	(98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152
2	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152
3	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152
4	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152
5	MIL-L-7808G	12.76	3.20	.9526	7.09-04	.152

L U B R I C A T I O N A N D F R I C T I O N D A T A

BEARING NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	INNER	ASPERITY FRICTION COEFFICIENT
1	.75	.3000-03	.1000-03	.10
2	.75	.1500-02	.5000-03	.10
3	1.50	.3000-02	.1000-02	.10
4	1.50	.3000-02	.1000-02	.10
5	1.50	.3000-02	.1000-02	.10

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F I T D A T A A N D M A T E R I A L P R O P E R T I E S

BEARING NUMBER	COLD FITS (MM TIGHT) SHAFT	HOUSING	EFFECTIVE WIDTHS SHAFT	INNER RING	OUTER RING	HOUSING
1	.0380	-.0076	50.0000	26.0000	26.0000	50.0000
2	.0203	-.0076	40.0000	26.0000	26.0000	40.0000
3	.0203	-.0076	26.0000	26.0000	26.0000	26.0000
4	.0203	-.0076	26.0000	26.0000	26.0000	26.0000
5	.0203	-.0076	40.0000	26.0000	26.0000	40.0000

BEARING NUMBER	EFFECTIVE DIAMETERS SHAFT I.D.	BEARING BORE	INNER RING AVE. O.D.	OUTER RING AVE. I.D.	BEARING O.D.	HOUSING O.D.
1	64.000	80.000	93.000	127.000	140.000	184.000
2	64.000	80.000	99.070	124.170	140.000	184.000
3	64.000	80.000	99.070	124.170	140.000	184.000
4	64.000	80.000	99.070	124.170	140.000	184.000
5	56.000	80.000	99.070	124.170	140.000	184.000

BEARING NUMBER (1)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
MODULUS OF ELASTICITY	204083.0	204083.0	204083.0	204083.0	41368.9
POISSONS RATIO	.3000	.3000	.3000	.3000	.3500
WEIGHT DENSITY	7.806	7.806	7.806	7.806	1.770
COEFF. OF THERMAL EXP.	.00001224	.00001224	.00001224	.00001224	.00002520

BEARING NUMBER (2)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
MODULUS OF ELASTICITY	204083.0	204083.0	204083.0	204083.0	41368.9
POISSONS RATIO	.3000	.3000	.3000	.3000	.3500
WEIGHT DENSITY	7.806	7.806	7.806	7.806	1.770
COEFF. OF THERMAL EXP.	.00001224	.00001224	.00001224	.00001224	.00002520

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

BEARING NUMBER (3)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
MODULUS OF ELASTICITY	204083.0	204083.0	204083.0	204083.0	41368.9
POISSONS RATIO	.3000	.3000	.3000	.3000	.3500
WEIGHT DENSITY	7.806	7.806	7.806	7.806	1.770
COEFF. OF THERMAL EXP.	.00001224	.00001224	.00001224	.00001224	.00002520

BEARING NUMBER (4)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
MODULUS OF ELASTICITY	204083.0	204083.0	204083.0	204083.0	41368.9
POISSONS RATIO	.3000	.3000	.3000	.3000	.3500
WEIGHT DENSITY	7.806	7.806	7.806	7.806	1.770
COEFF. OF THERMAL EXP.	.00001224	.00001224	.00001224	.00001224	.00002520

BEARING NUMBER (5)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
MODULUS OF ELASTICITY	204083.0	204083.0	204083.0	204083.0	41368.9
POISSONS RATIO	.3000	.3000	.3000	.3000	.3500
WEIGHT DENSITY	7.806	7.806	7.806	7.806	1.770
COEFF. OF THERMAL EXP.	.00001224	.00001224	.00001224	.00001224	.00002520

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

STEADY STATE TEMPERATURE CALCULATION. ITERATION LIMIT 10. ABSOLUTE ACCURACY 1.00 DEGREES
INTERMEDIATE OUTPUT WILL BE OBTAINED

UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

NODE POINTERS

BRG	SHAFT	I. RING	I. RACE	ROLL EL.	O. RACE	O. RING	HOUSING	BULK	FLANGE
1	4	5	5	6	7	7	8	41	0
2	13	14	14	15	16	16	17	43	0
3	19	20	20	21	22	22	23	46	0
4	24	25	25	26	27	27	28	49	0
5	31	32	32	33	34	34	35	52	0

NODES WHERE BEARING HEAT IS GENERATED

BRG	INNER RACE		OUTER RACE		CAGE		DRAG		FLANGE	
1	5	6	6	7	7	7	6	41	0	0
2	14	15	15	16	14	14	15	43	0	0
3	20	21	21	22	20	20	21	46	0	0
4	25	26	26	27	25	25	26	49	0	0
5	32	33	33	34	32	32	33	52	0	0

CONSTANT GENERATED HEATS

NODE	GEN. HEAT	NODE	GEN. HEAT	NODE	GEN. HEAT	NODE	GEN. HEAT	NODE	GEN. HEAT
10	2500.00	37	100.00	58	2500.00				

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

HEAT TRANSFER COEFFICIENTS

TYPE	INDEX	COEFFICIENTS				
CONDUCTION	1	53.6517				
CONDUCTION	2	46.7293				
CONDUCTION	3	50.8231				
FORCED CONVECTION	21	.300000	.570000	.000000	.180000-01	34.0000
		.170000	.000000	900.000	2000.00	25.0000
		.180000-01	218.330	.700000-03	103.434	.421460+11
		-4.03921	.900000-03	-.570000		
FORCED CONVECTION	22	.300000	.570000	.000000	.250000-01	48.0000
		.280000-01	.220000-04	1.21000	1000.00	.000000
		.000000	.000000	.000000	187.707	.000000
		.000000	.000000	.000000		
FORCED CONVECTION	23	37.0000				
FORCED CONVECTION	24	5000.00				
FORCED CONVECTION	25	.270000-01	.800000	.333000	.500000-03	40.0000
		.170000	.000000	900.000	2000.00	25.0000
		.180000-01	218.330	.700000-03	2101.67	.421460+11
		-4.03921	.900000-03	-.467000		
FORCED CONVECTION	26	.300000	.570000	.000000	.250000-01	48.0000
		.170000	.000000	900.000	2000.00	25.0000
		.180000-01	218.330	.700000-03	109.315	.421460+11
		-4.03921	.900000-03	-.570000		
FLUID FLOW	41	28.0000				
FLUID FLOW	42	42.0000				
FLUID FLOW	43	21.0000				
FLUID FLOW	44	63.0000				

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

HEAT TRANSFER COEFFICIENTS

TYPE	INDEX	COEFFICIENTS
FLUID FLOW	45	105.000
FLUID FLOW	46	147.000
FLUID FLOW	47	337.000
FLUID FLOW	48	136.000

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER

ALL LENGTHS ARE IN MILLIMETERS. A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

TYPE OF HEAT TR.	INDEX		NODE		NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
CONDUCTION	1	BETWEEN	1	AND	2	50.0000	24.0000	44.0000
CONDUCTION	-1	BETWEEN	2	AND	3	20.0000	140.0000	150.0000
CONDUCTION	1	BETWEEN	2	AND	8	77.0000	26.0000	84.0000
CONDUCTION	1	BETWEEN	8	AND	12	95.0000	5.0000	70.0000
CONDUCTION	1	BETWEEN	12	AND	18	100.0000	6.0000	74.0000
CONDUCTION	1	BETWEEN	18	AND	29	104.0000	20.0000	53.0000
CONDUCTION	-1	BETWEEN	29	AND	30	20.0000	140.0000	150.0000
CONDUCTION	1	BETWEEN	29	AND	36	104.0000	22.0000	26.0000
CONDUCTION	1	BETWEEN	29	AND	28	92.0000	26.0000	22.0000
CONDUCTION	1	BETWEEN	36	AND	35	92.0000	26.0000	22.0000
CONDUCTION	1	BETWEEN	36	AND	39	104.0000	22.0000	26.0000
CONDUCTION	1	BETWEEN	39	AND	38	92.0000	26.0000	22.0000
CONDUCTION	1	BETWEEN	3	AND	30	170.0000	6.0000	276.0000
CONDUCTION	1	BETWEEN	17	AND	23	81.0000	22.0000	26.0000
CONDUCTION	1	BETWEEN	23	AND	28	81.0000	22.0000	26.0000
CONDUCTION	1	BETWEEN	28	AND	35	81.0000	22.0000	26.0000
CONDUCTION	1	BETWEEN	35	AND	38	81.0000	22.0000	26.0000
CONDUCTION	2	BETWEEN	8	AND	7	70.0000	26.0000	10.0000
CONDUCTION	2	BETWEEN	17	AND	16	74.0000	26.0000	13.0000
CONDUCTION	2	BETWEEN	23	AND	22	74.0000	26.0000	13.0000
CONDUCTION	2	BETWEEN	28	AND	27	74.0000	26.0000	13.0000
CONDUCTION	2	BETWEEN	35	AND	34	74.0000	26.0000	13.0000
CONDUCTION	3	BETWEEN	4	AND	5	40.0000	26.0000	7.0000
CONDUCTION	3	BETWEEN	4	AND	9	36.0000	8.0000	36.0000
CONDUCTION	3	BETWEEN	9	AND	10	56.0000	12.0000	40.0000

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DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER

ALL LENGTHS ARE IN MILLIMETERS. A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

TYPE OF HEAT TR.	INDEX		NODE		NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
CONDUCTION	3	BETWEEN	10	AND	11	58.0000	14.0000	44.0000
CONDUCTION	3	BETWEEN	5	AND	9	45.0000	3.0000	36.0000
CONDUCTION	3	BETWEEN	11	AND	14	42.0000	5.0000	36.0000
CONDUCTION	3	BETWEEN	11	AND	13	38.0000	7.0000	36.0000
CONDUCTION	3	BETWEEN	13	AND	14	40.0000	26.0000	6.0000
CONDUCTION	3	BETWEEN	13	AND	19	35.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	14	AND	20	43.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	19	AND	20	40.0000	26.0000	8.0000
CONDUCTION	3	BETWEEN	16	AND	22	66.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	22	AND	27	66.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	20	AND	25	43.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	19	AND	24	36.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	24	AND	25	40.0000	26.0000	8.0000
CONDUCTION	3	BETWEEN	27	AND	34	66.0000	8.0000	25.0000
CONDUCTION	3	BETWEEN	25	AND	32	43.0000	3.0000	26.0000
CONDUCTION	3	BETWEEN	24	AND	31	35.0000	8.0000	26.0000
CONDUCTION	3	BETWEEN	31	AND	32	40.0000	26.0000	8.0000
CONDUCTION	3	BETWEEN	31	AND	37	40.0000	8.0000	40.0000
CONDUCTION	3	BETWEEN	32	AND	37	46.0000	6.0000	32.0000
FORCED CONVECTION	21	BETWEEN	5	AND	41	45.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	6	AND	41	17.0000	220.0000	
FORCED CONVECTION	21	BETWEEN	7	AND	41	63.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	14	AND	43	46.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	15	AND	43	142.5000	19.0000	
FORCED CONVECTION	21	BETWEEN	16	AND	43	63.0000	26.0000	

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DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER

ALL LENGTHS ARE IN MILLIMETERS, A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

TYPE OF HEAT TR.	INDEX		NODE		NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
FORCED CONVECTION	21	BETWEEN	20	AND	46	46.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	21	AND	46	142.5000	19.0000	
FORCED CONVECTION	21	BETWEEN	22	AND	45	63.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	25	AND	49	46.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	26	AND	49	142.5000	19.0000	
FORCED CONVECTION	21	BETWEEN	27	AND	49	63.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	32	AND	52	46.0000	26.0000	
FORCED CONVECTION	21	BETWEEN	33	AND	52	142.5000	19.0000	
FORCED CONVECTION	21	BETWEEN	34	AND	52	63.0000	25.0000	
FORCED CONVECTION	22	BETWEEN	1	AND	40	46.0000	23.0000	
FORCED CONVECTION	22	BETWEEN	2	AND	40	60.0000	20.0000	
FORCED CONVECTION	22	BETWEEN	3	AND	40	70.0000	140.0000	
FORCED CONVECTION	22	BETWEEN	4	AND	40	40.0000	60.0000	
FORCED CONVECTION	22	BETWEEN	8	AND	40	50.0000	40.0000	
FORCED CONVECTION	22	BETWEEN	9	AND	40	40.0000	40.0000	
FORCED CONVECTION	22	BETWEEN	10	AND	40	65.0000	100.0000	
FORCED CONVECTION	22	BETWEEN	11	AND	40	42.0000	30.0000	
FORCED CONVECTION	22	BETWEEN	12	AND	40	95.0000	100.0000	
FORCED CONVECTION	22	BETWEEN	17	AND	40	80.0000	40.0000	
FORCED CONVECTION	22	BETWEEN	18	AND	40	110.0000	20.0000	
FORCED CONVECTION	22	BETWEEN	29	AND	40	120.0000	10.0000	
FORCED CONVECTION	22	BETWEEN	30	AND	40	80.0000	130.0000	
FORCED CONVECTION	23	BETWEEN	1	AND	50	46.0000	23.0000	
FORCED CONVECTION	25	BETWEEN	42	AND	13	32.0000	5.0000	
FORCED CONVECTION	25	BETWEEN	45	AND	19	32.0000	5.0000	

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER

ALL LENGTHS ARE IN MILLIMETERS. A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

TYPE OF HEAT TR.	INDEX		NODE		NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
FORCED CONVECTION	25	BETWEEN	48	AND	24	32.0000	5.0000	
FORCED CONVECTION	25	BETWEEN	51	AND	31	32.0000	5.0000	
FORCED CONVECTION	26	BETWEEN	10	AND	58	65.0000	100.0000	
FORCED CONVECTION	23	BETWEEN	2	AND	60	80.0000	20.0000	
FORCED CONVECTION	23	BETWEEN	3	AND	60	70.0000	160.0000	
FORCED CONVECTION	23	BETWEEN	29	AND	60	120.0000	10.0000	
FORCED CONVECTION	23	BETWEEN	30	AND	60	80.0000	160.0000	
FORCED CONVECTION	23	BETWEEN	35	AND	60	110.0000	30.0000	
FORCED CONVECTION	23	BETWEEN	38	AND	60	82.0000	20.0000	
FORCED CONVECTION	23	BETWEEN	39	AND	60	110.0000	45.0000	
FORCED CONVECTION	23	BETWEEN	12	AND	60	110.0000	50.0000	
FORCED CONVECTION	23	BETWEEN	8	AND	60	100.0000	60.0000	
FORCED CONVECTION	23	BETWEEN	18	AND	60	50.0000	25.0000	
FORCED CONVECTION	24	BETWEEN	12	AND	57	95.0000	100.0000	
FLUID FLOW	41	FROM	61	TO	41	(INDEX 41)		
FLUID FLOW	41	FROM	41	TO	57	(INDEX 47)		
FLUID FLOW	42	FROM	61	TO	42	(INDEX 42)		
FLUID FLOW	42	FROM	61	TO	45	(INDEX 42)		
FLUID FLOW	42	FROM	61	TO	48	(INDEX 42)		
FLUID FLOW	42	FROM	61	TO	51	(INDEX 42)		
FLUID FLOW	42	FROM	42	TO	43	(INDEX 42)		
FLUID FLOW	42	FROM	45	TO	46	(INDEX 42)		
FLUID FLOW	42	FROM	48	TO	49	(INDEX 42)		
FLUID FLOW	42	FROM	51	TO	52	(INDEX 42)		
FLUID FLOW	42	FROM	43	TO	44	(INDEX 43)		

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER

ALL LENGTHS ARE IN MILLIMETERS, A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

TYPE OF HEAT TR.	INDEX		NODE		NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
FLUID FLOW	43	FROM	43	TO	57	(INDEX 47)		
FLUID FLOW	42	FROM	46	TO	44	(INDEX 43)		
FLUID FLOW	42	FROM	45	TO	47	(INDEX 43)		
FLUID FLOW	42	FROM	49	TO	47	(INDEX 43)		
FLUID FLOW	42	FROM	49	TO	50	(INDEX 43)		
FLUID FLOW	42	FROM	52	TO	50	(INDEX 43)		
FLUID FLOW	42	FROM	52	TO	53	(INDEX 43)		
FLUID FLOW	42	FROM	47	TO	55	(INDEX 45)		
FLUID FLOW	43	FROM	53	TO	54	(INDEX 44)		
FLUID FLOW	42	FROM	50	TO	54	(INDEX 44)		
FLUID FLOW	44	FROM	54	TO	55	(INDEX 45)		
FLUID FLOW	42	FROM	44	TO	56	(INDEX 45)		
FLUID FLOW	45	FROM	55	TO	56	(INDEX 46)		
FLUID FLOW	46	FROM	56	TO	57	(INDEX 47)		
FLUID FLOW	47	FROM	57	TO	61	(INDEX 47)		
FLUID FLOW	48	FROM	61	TO	58	(INDEX 48)		
FLUID FLOW	48	FROM	58	TO	57	(INDEX 47)		
BEARING CONDUCTION	51	BETWEEN	6	AND	5	1.0000	1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	6	AND	7	1.0000	-1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	15	AND	14	2.0000	1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	15	AND	16	2.0000	-1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	21	AND	20	3.0000	1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	21	AND	22	3.0000	-1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	26	AND	25	4.0000	1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	26	AND	27	4.0000	-1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	33	AND	32	5.0000	1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	33	AND	34	5.0000	-1.0000	1.0000

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T E M P E R A T U R E M A P

TEMPERATURES ARE IN DEGREES CELSIUS. THE FIRST 58 TEMPERATURES ARE CALCULATED, THE OTHERS ARE KNOWN

STEADY STATE TEMPERATURE CALCULATION. INITIAL TEMPERATURES

CALCULATED TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
1	75.000	2	75.000	3	75.000	4	75.000	5	75.000
6	75.000	7	75.000	8	75.000	9	75.000	10	75.000
11	75.000	12	75.000	13	75.000	14	75.000	15	75.000
16	75.000	17	75.000	18	75.000	19	75.000	20	75.000
21	75.000	22	75.000	23	75.000	24	75.000	25	75.000
26	75.000	27	75.000	28	75.000	29	75.000	30	75.000
31	75.000	32	75.000	33	75.000	34	75.000	35	75.000
36	75.000	37	75.000	38	75.000	39	75.000	40	75.000
41	75.000	42	75.000	43	75.000	44	75.000	45	75.000
46	75.000	47	75.000	48	75.000	49	75.000	50	75.000
51	75.000	52	75.000	53	75.000	54	75.000	55	75.000
56	75.000	57	75.000	58	75.000				

KNOWN BOUNDARY TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
59	75.000	60	24.000	61	70.000				

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SHAFT GEOMETRY. BEARING LOCATIONS AND SHAFT LOAD. PLANE X - Y.

12 GEOMETRIC SECTIONS 1 LOAD SECTION(S). 5 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT		BEARING SEAT POS.ERR	BEARING SEAT DEFL/FOR	BEARING SEAT ANG.ERR	BEARING SEAT DEFL/MOM
1	.0	.0	65.4	.0	86.0					1				
2	13.0	55.4	65.4	85.0	86.0					2	.000	0.00	.0000	0.00
3	33.0	65.4	65.4	85.0	84.0					3				
4	40.0	55.4	55.4	84.0	84.0					4				
5	49.0	76.0	76.0	100.0	122.0					5				
6	58.0	92.0	92.0	124.0	124.0					6				
7	78.0	92.0	92.0	130.5	130.5	3900.0	318500.0			7				
8	104.0	92.0	92.0	139.0	139.0					8				
9	109.0	86.0	86.0	141.0	141.0					9				
10	112.0	78.0	78.0	96.0	96.0					10				
11	119.0	66.0	66.0	84.0	84.0					11				
12	122.0	64.0	64.0	84.0	84.0					12				
13	128.0	64.0	64.0	84.0	90.0					13				
14	148.0	64.0	64.0	90.0	90.0					14	.000	0.00	.0000	0.00
15	174.0	64.0	64.0	90.0	90.0					15	.000	0.00	.0000	0.00
16	200.0	64.0	64.0	90.0	90.0					16	.000	0.00	.0000	0.00
17	226.0	64.0	64.0	90.0	90.0					17	.000	0.00	.0000	0.00
18	239.0	64.0	.0	90.0	.0					18				

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD. PLANE X - Z.

12 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 5 BEARINGS. MODULUS OF ELASTICITY = 2.041+05

THRUST LOAD = 4.900+03

	POSI-TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT		BEARING SEAT POS.ERR	BEARING SEAT DEFL/FOR	BEARING SEAT ANG.ERR	BEARING SEAT DEFL/MOM
1	.0	.0	65.4	.0	86.0					1				
2	13.0	65.4	65.4	86.0	86.0					2	.000	0.00	.0000	0.00
3	33.0	65.4	65.4	86.0	84.0					3				
4	40.0	65.4	65.4	84.0	84.0					4				
5	49.0	76.0	76.0	100.0	122.0					5				
6	58.0	92.0	92.0	124.0	124.0					6				
7	78.0	92.0	92.0	130.5	130.5	11500.0				7				
8	104.0	92.0	92.0	139.0	139.0					8				
9	109.0	86.0	86.0	141.0	141.0					9				
10	112.0	78.0	78.0	96.0	96.0					10				
11	117.0	66.0	66.0	84.0	84.0					11				
12	122.0	64.0	64.0	84.0	84.0					12				
13	128.0	64.0	64.0	84.0	90.0					13				
14	148.0	64.0	64.0	90.0	90.0					14	.000	0.00	.0000	0.00
15	174.0	64.0	64.0	90.0	90.0					15	.000	0.00	.0000	0.00
16	200.0	64.0	64.0	90.0	90.0					16	.000	0.00	.0000	0.00
17	226.0	64.0	64.0	90.0	90.0					17	.000	0.00	.0000	0.00
18	239.0	64.0	.0	90.0	.0					18				

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

BEARING SYSTEM OUTPUT METRIC UNITS

LINEAR (MM) AND ANGULAR (RADIANS) DEFLECTIONS — REACTION FORCES (N) AND MOMENTS (MM-N)

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	2.581-03	5.993-04	9.589-03	-3.579-05	5.621-05	0.000	459.	6.802+03	-319.	355.
2	2.581-03	7.524-03	1.205-02	3.314-06	3.545-05	2.389+03	670.	1.252+03	3.557+04	-1.834+04
3	2.581-03	8.370-03	1.188-02	8.399-06	3.025-05	2.380+03	769.	1.243+03	3.543+04	-2.124+04
4	2.581-03	9.109-03	1.152-02	1.129-05	2.709-05	2.410+03	868.	1.245+03	3.547+04	-2.405+04
5	2.581-03	9.780-03	1.129-02	1.370-05	2.472-05	-2.279+03	1.079+03	1.117+03	-3.104+04	3.071+04

	FATIGUE LIFE (HOURS)			H/SIGMA		LUBE-LIFE FACTOR		MATERIAL FACTOR	
BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	O. RACE	I. RACE	O. RACE	I. RACE
1	2.594+05	2.009+05	1.221+05	1.05	.814	.455	.432	5.00	5.00
2	7.958+04	1.575+05	5.632+04	1.85	1.74	.639	.602	5.00	5.00
3	7.563+04	1.492+05	5.350+04	1.76	1.68	.611	.586	5.00	5.00
4	7.380+04	1.376+05	5.122+04	1.75	1.66	.607	.579	5.00	5.00
5	7.605+04	1.418+05	5.279+04	1.75	1.61	.607	.565	5.00	5.00

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CENTIGRADE)

BRG.	SHAFT	I. RING	I. RACE	I. FLNG.	ROLL. EL.	O. FLNG.	O. RACE	O. RING	HSG.	BULK LUBE
1	116.	119.	118.	118.	115.	107.	107.	107.	103.	102.
2	78.7	82.3	82.3	82.3	86.5	85.0	85.0	85.0	86.2	78.8
3	79.6	84.6	84.6	84.6	94.3	88.1	88.1	88.1	87.4	84.8
4	80.1	85.3	85.3	85.3	94.6	88.4	88.4	88.4	87.3	85.1
5	82.0	87.3	87.3	87.3	95.3	88.4	88.4	88.4	86.3	86.0

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B E A R I N G S Y S T E M O U T P U T METRIC UNITS

FRICTIONAL HEAT GENERATION RATE (WATTS) AND FRICTION TORQUE (N-MM)

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	1.033+03	0.000	405.	0.000	625.	20.5	10.4	2.096+03	1.472+03
2	266.	0.000	160.	0.000	293.	36.6	38.6	794.	557.
3	303.	0.000	194.	0.000	555.	32.1	34.3	1.118+03	785.
4	301.	0.000	195.	0.000	553.	31.5	34.1	1.115+03	783.
5	301.	0.000	190.	0.000	549.	31.1	33.5	1.104+03	776.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (MM)		CONDUCTIVITY (W/DEG.C)	
1	.258	.207	.998	.969	.915	.932	.767	.225	18.5	9.18
2	.354	.332	.999	.993	.867	.869	1.16	.527	7.12	2.98
3	.337	.321	1.000	.998	.874	.875	1.69	.813	7.41	3.05
4	.335	.317	1.000	.998	.875	.876	1.69	.814	7.45	3.10
5	.335	.307	1.000	.998	.875	.880	1.69	.815	7.40	3.08

FIT PRESSURES (N/MM2) BEARING CLEARANCES (MM) SPEED GIVING ZERO FIT PRESSURE

BRG.	SHAFT-COLD,	OPER.	HSG.-COLD,	OPER.	ORIGINAL	CHANGE	OPERATING	SHAFT-INNER RING (RPM)
1	10.5	8.41	0.000	0.000	3.800-02	-4.339-02	-5.390-03	3.390+04
2	6.63	3.37	0.000	0.000	2.053-02	-1.169-02	1.061-02	2.113+04
3	5.57	2.42	0.000	0.000	2.053-02	-1.126-02	1.104-02	2.021+04
4	5.57	2.38	0.000	0.000	2.053-02	-1.154-02	1.076-02	2.012+04
5	7.67	2.75	0.000	0.000	2.053-02	-1.500-02	7.232-03	1.851+04

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B E A R I N G   S Y S T E M   O U T P U T   METRIC UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM3)	VISCOSITY KINEMATIC (CS)	VISCOSITY DYNAMIC (CP)	PRESSURE VISCOSITY COEFFICIENT (MM2/N)
BRG. 1	OUTER	107.085	.8877	2.827	2.509	.9817-02
	INNER	117.977	.8900	2.433	2.141	.9235-02
	BULK	102.361	.8910	3.032	2.702	.1008-01
BRG. 2	OUTER	84.962	.9034	4.052	3.660	.1114-01
	INNER	82.322	.9052	4.255	3.852	.1131-01
	BULK	78.787	.9078	4.552	4.132	.1155-01
BRG. 3	OUTER	88.094	.9012	3.831	3.452	.1094-01
	INNER	84.629	.9036	4.077	3.684	.1116-01
	BULK	84.752	.9035	4.068	3.675	.1115-01
BRG. 4	OUTER	88.408	.9009	3.810	3.432	.1092-01
	INNER	85.257	.9032	4.030	3.640	.1112-01
	BULK	85.083	.9033	4.043	3.652	.1113-01
BRG. 5	OUTER	88.360	.9010	3.813	3.435	.1092-01
	INNER	87.273	.9017	3.887	3.505	.1099-01
	BULK	86.030	.9026	3.974	3.567	.1107-01

C A G E   D A T A   METRIC UNITS

CAGE RAIL = RING LAND DATA   CAGE SPEED DATA

BRG.	TORQUE (MM-N)	HEAT RATE (WATTS)	SEP.FORCE (NEWTONS)	ECCENTRICITY RATIO	EPICYCLIC SPEED (RAD/SEC)	EPICYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)	CALC/EPIC RATIO	CAGE/SHAFT RATIO
1	-17.2	10.4	0.000	0.000	602.	5.749+03	602.	5.749+03	1.000	.423
2	49.1	38.6	1.618-03	1.000-02	638.	6.093+03	638.	6.093+03	1.00	.449
3	43.7	34.3	1.439-03	1.000-02	638.	6.097+03	638.	6.097+03	1.00	.448
4	43.4	34.1	1.430-03	1.000-02	638.	6.095+03	638.	6.095+03	1.00	.448
5	42.6	33.5	1.405-03	1.000-02	638.	6.095+03	638.	6.095+03	1.00	.448

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R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-4497.618	-.115	.000	4497.618	602.043	-180.00	180.00
22.50	-4497.618	-.143	.000	4497.618	602.043	-180.00	180.00
45.00	-4497.618	-.093	.000	4497.618	602.043	-180.00	180.00
67.50	-4497.618	-.076	.000	4497.618	602.043	-180.00	180.00
90.00	-4497.618	-.069	.000	4497.618	602.043	-180.00	180.00
112.50	-4497.618	-.017	.000	4497.618	602.043	-180.00	180.00
135.00	-4497.618	.021	.000	4497.618	602.043	180.00	180.00
157.50	-4497.618	.083	.000	4497.618	602.043	180.00	180.00
180.00	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
202.50	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
225.00	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
247.50	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
270.00	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
292.50	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
315.00	-4497.618	.000	.000	4497.618	602.043	180.00	180.00
337.50	-4497.618	.000	.000	4497.618	602.043	180.00	180.00

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R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	.000	310.490	139.425	549.366	292.364	.3104	.6935	.00	.00
22.50	.000	1394.194	721.827	716.860	611.830	.2746	.4611	.00	.00
45.00	.000	1327.221	1255.931	826.604	802.054	.2555	.3835	.00	.20
67.50	.000	2279.878	1629.949	886.638	905.016	.2465	.3537	.00	.00
90.00	-.000	2404.839	1733.833	902.574	925.105	.2442	.3558	.00	.00
112.50	-.000	2208.612	1537.008	867.997	858.406	.2492	.3589	.00	.00
135.00	-.000	1763.491	1092.359	792.258	733.966	.2613	.4113	.00	.00
157.50	-.000	1208.142	538.345	563.445	524.562	.2849	.5061	.00	.00
180.00	-.000	671.137	.000	489.194	.000	.3258	.0000	.00	.00
202.50	-.000	671.050	.000	489.162	.000	.3259	.0000	.00	.00
225.00	-.000	671.050	.000	489.162	.000	.3258	.0000	.00	.00
247.50	-.000	671.050	.000	489.162	.000	.3258	.0000	.00	.00
270.00	-.000	671.050	.000	489.162	.000	.3259	.0000	.00	.00
292.50	-.000	671.050	.000	489.162	.000	.3258	.0000	.00	.00
315.00	-.000	671.050	.000	489.162	.000	.3258	.0000	.00	.00
337.50	-.000	671.050	.000	489.162	.000	.3258	.0000	.00	.00

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-4157.256	729.759	-4.559	4220.823	624.693	170.04	-179.94
24.00	-4092.231	815.425	-5.046	4172.380	618.054	168.72	-179.93
48.00	-4057.495	866.523	-5.327	4148.995	614.807	167.94	-179.92
72.00	-4055.935	871.985	-5.361	4148.672	614.789	167.87	-179.92
96.00	-4087.756	832.493	-5.144	4171.667	617.959	168.49	-179.93
120.00	-4150.799	733.940	-4.708	4218.718	624.494	169.71	-179.93
144.00	-4235.714	652.725	-4.138	4285.714	633.924	171.24	-179.94
168.00	-4326.513	549.807	-3.545	4361.309	644.634	172.76	-179.93
192.00	-4405.380	462.386	-3.027	4430.575	654.648	174.01	-179.36
216.00	-4461.433	402.038	-2.660	4479.512	661.715	174.85	-179.97
240.00	-4481.669	376.864	-2.504	4497.486	664.311	175.19	-179.97
264.00	-4463.934	399.766	-2.580	4480.978	661.904	175.01	-179.97
288.00	-4411.415	439.990	-2.832	4433.304	654.992	174.30	-179.96
312.00	-4333.271	521.645	-3.365	4364.557	645.084	173.14	-179.96
336.00	-4242.902	624.210	-3.959	4288.575	634.236	171.63	-179.95

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-1.137	939.437	366.464	1108.148	934.248	.0403	.0590	11.67	31.24
24.00	-.493	1033.750	475.546	1144.056	1084.464	.0418	.0578	13.22	29.85
48.00	-.141	1106.363	555.072	1170.239	1141.828	.0426	.0572	14.13	29.11
72.00	.133	1111.429	560.478	1172.022	1145.523	.0426	.0571	14.22	29.16
96.00	.470	1045.257	489.566	1148.654	1034.272	.0419	.0577	13.49	29.97
120.00	1.103	954.615	382.946	1114.084	1098.935	.0404	.0598	12.07	31.42
144.00	2.686	879.630	286.328	1083.728	915.740	.0385	.0604	10.27	33.19
168.00	2.239	830.971	213.933	1063.744	831.019	.0366	.0624	8.43	34.33
192.00	.920	805.959	165.379	1052.963	752.625	.0350	.0644	7.03	36.33
216.00	.361	794.514	138.197	1047.954	718.319	.0335	.0660	6.04	37.23
240.00	.006	790.222	127.390	1046.064	699.085	.0330	.0665	5.64	37.50
264.00	-.347	790.853	133.736	1045.342	710.610	.0335	.0661	5.85	37.14
288.00	-.896	798.670	157.610	1049.778	750.490	.0349	.0647	6.68	36.18
312.00	-2.136	820.347	202.464	1059.191	815.831	.0365	.0627	8.05	34.73
336.00	-2.792	864.848	270.781	1078.008	898.856	.0384	.0607	9.81	32.99

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 3 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-4145.807	745.555	-4.649	4212.314	623.512	169.81	-179.94
24.00	-4082.832	830.625	-5.125	4166.442	617.180	168.50	-179.93
48.00	-4051.950	875.703	-5.380	4145.503	614.344	167.80	-179.92
72.00	-4055.339	874.166	-5.374	4148.490	614.775	167.84	-179.92
96.00	-4082.940	826.370	-5.111	4175.532	618.436	168.59	-179.93
120.00	-4162.112	739.753	-4.629	4227.345	625.639	169.92	-179.94
144.00	-4251.739	633.001	-4.024	4298.603	635.747	171.53	-179.95
168.00	-4345.053	527.339	-3.412	4376.945	646.928	173.08	-179.95
192.00	-4424.527	440.488	-2.894	4446.400	656.924	174.31	-179.96
216.00	-4476.445	383.461	-2.545	4492.840	663.640	175.10	-179.97
240.00	-4491.309	363.265	-2.418	4506.575	665.623	175.38	-179.97
264.00	-4468.209	382.908	-2.536	4484.577	662.419	175.10	-179.97
288.00	-4409.365	440.545	-2.884	4431.318	654.700	174.29	-179.96
312.00	-4325.977	529.303	-3.410	4358.239	644.176	173.02	-179.95
336.00	-4232.131	637.509	-4.036	4279.929	633.013	171.43	-179.95

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 3 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.881	953.357	333.103	1113.595	1009.074	.0497	.0569	11.95	31.02
24.00	-.374	1051.670	495.561	1150.628	1093.470	.0514	.0555	13.49	29.67
48.00	-.099	1119.976	569.970	1175.019	1151.953	.0522	.0647	14.29	29.02
72.00	.153	1113.026	562.317	1172.583	1146.775	.0521	.0648	14.25	29.17
96.00	.459	1036.771	478.141	1145.169	1036.433	.0511	.0656	13.39	30.11
120.00	1.090	941.507	367.442	1109.951	935.132	.0493	.0672	11.82	31.66
144.00	2.488	867.491	270.842	1079.104	878.924	.0467	.0694	9.93	33.49
168.00	1.710	823.458	201.421	1060.529	814.428	.0441	.0719	8.12	35.24
192.00	.720	891.272	155.231	1050.917	746.636	.0419	.0743	6.57	36.61
216.00	.270	791.714	129.515	1046.722	702.950	.0406	.0760	5.74	37.43
240.00	-.036	788.503	122.031	1045.305	693.140	.0402	.0766	5.42	37.63
264.00	-.360	799.969	131.479	1045.952	705.485	.0403	.0758	5.74	37.17
288.00	-.893	799.006	158.279	1049.926	751.552	.0424	.0740	6.69	36.12
312.00	-2.133	823.302	207.423	1060.462	822.439	.0447	.0715	8.18	34.59
336.00	-2.123	872.427	281.059	1081.148	910.098	.0473	.0690	10.04	32.78

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 4 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-4131.371	765.431	-4.761	4201.682	622.041	168.50	-179.93
24.00	-4071.286	847.645	-5.222	4158.594	615.116	168.24	-179.93
48.00	-4044.746	887.042	-5.444	4140.875	613.727	167.63	-179.92
72.00	-4052.526	879.221	-5.403	4146.809	614.556	167.76	-179.92
96.00	-4094.892	824.709	-5.103	4177.108	619.719	168.61	-179.93
120.00	-4168.807	731.303	-4.585	4232.570	626.432	170.04	-179.94
144.00	-4261.743	621.306	-3.957	4306.796	636.910	171.71	-179.95
168.00	-4356.063	514.741	-3.337	4386.377	648.230	173.26	-179.96
192.00	-4434.212	429.304	-2.825	4454.946	658.157	174.47	-179.96
216.00	-4482.891	375.578	-2.496	4498.597	654.472	175.21	-179.97
240.00	-4493.250	360.741	-2.402	4507.709	665.785	175.41	-179.97
264.00	-4464.139	385.245	-2.556	4480.789	661.858	175.05	-179.97
288.00	-4400.283	450.012	-2.941	4423.235	653.532	174.16	-179.96
312.00	-4312.725	544.691	-3.500	4346.988	642.566	172.80	-179.95
336.00	-4217.164	656.545	-4.145	4267.967	631.325	171.15	-179.94

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 4 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.775	372.367	405.483	1120.348	1028.352	.0511	.0693	12.30	30.73
24.00	-.325	1075.177	521.551	1159.138	1118.364	.0527	.0679	13.78	29.44
48.00	-.057	1139.245	591.409	1181.719	1166.219	.0534	.0671	14.50	28.88
72.00	.193	1120.253	570.595	1175.116	1152.374	.0532	.0673	14.35	29.13
96.00	.514	1033.236	474.339	1143.866	1093.545	.0520	.0683	13.35	30.17
120.00	1.185	934.246	358.868	1106.103	987.330	.0500	.0702	11.67	31.81
144.00	2.478	861.309	262.340	1076.531	889.418	.0473	.0726	9.73	33.69
168.00	1.491	819.498	194.393	1058.826	804.843	.0446	.0753	7.90	35.42
192.00	.638	799.179	150.072	1050.302	738.330	.0425	.0779	6.49	36.76
216.00	.225	790.955	127.007	1046.344	698.382	.0412	.0795	5.62	37.52
240.00	-.073	788.294	120.474	1045.213	686.197	.0410	.0800	5.38	37.63
264.00	-.407	790.584	132.496	1046.224	709.302	.0417	.0790	5.80	37.09
288.00	-.981	801.382	163.182	1050.965	759.233	.0435	.0769	6.85	35.94
312.00	-2.244	824.793	215.660	1062.814	833.184	.0459	.0742	8.44	34.35
336.00	-1.827	883.910	296.330	1085.870	926.281	.0487	.0715	10.38	32.50

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

R O L L I N G   E L E M E N T   O U T P U T   FOR BEARING NUMBER 5 METRIC UNITS

AZIMUTH	ANGULAR SPEEDS (RADIANS/SECOND)				SPEED VECTOR ANGLES (DEGREES)		
ANGLE (DEG.)	WX	WY	WZ	TOTAL	ORBITAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-4109.180	-786.567	4.973	4183.787	619.509	-169.16	179.93
24.00	-4058.059	-854.672	5.252	4147.095	614.452	-168.11	179.93
48.00	-4042.305	-874.342	5.359	4135.786	612.908	-167.80	179.92
72.00	-4061.716	-843.495	5.197	4143.461	614.506	-168.26	179.93
96.00	-4116.435	-765.722	4.747	4187.051	619.439	-169.46	179.93
120.00	-4200.041	-654.650	4.116	4250.756	628.783	-171.14	179.94
144.00	-4297.467	-535.014	3.425	4330.643	640.126	-172.90	179.95
168.00	-4390.275	-429.533	2.799	4411.238	651.711	-174.41	179.96
192.00	-4462.078	-353.696	2.338	4476.075	661.104	-175.47	179.97
216.00	-4500.408	-316.319	2.112	4511.553	666.268	-175.97	179.97
240.00	-4498.903	-322.719	2.150	4510.463	666.117	-175.90	179.97
264.00	-4457.174	-370.712	2.449	4472.565	660.521	-175.25	179.97
288.00	-4382.806	-454.362	2.958	4406.296	651.046	-174.08	179.96
312.00	-4288.106	-563.787	3.605	4325.011	639.390	-172.51	179.95
336.00	-4190.714	-681.961	4.284	4245.842	628.170	-170.76	179.94

POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

ROLLING ELEMENT OUTPUT FOR BEARING NUMBER 5 METRIC UNITS

AZIMUTH	NORMAL FORCES (NEWTONS)			HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
ANGLE (DEG.)	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	-.603	1001.440	439.386	1132.010	1056.249	.0516	.0777	-12.70	-30.08
24.00	-.237	1099.944	543.964	1167.971	1137.634	.0523	.0750	-13.94	-28.87
48.00	.007	1138.883	591.175	1181.596	1166.066	.0533	.0755	-14.30	-28.43
72.00	.254	1088.123	535.969	1163.772	1128.576	.0529	.0752	-13.75	-28.82
96.00	.634	984.538	421.227	1125.605	1041.493	.0514	.0780	-12.35	-30.00
120.00	1.479	887.655	304.463	1087.402	934.679	.0491	.0807	-10.37	-31.74
144.00	2.427	824.399	215.543	1061.193	833.033	.0463	.0841	-8.32	-33.63
168.00	1.158	793.150	156.015	1047.354	747.351	.0436	.0875	-6.55	-35.33
192.00	.432	780.512	120.871	1041.762	696.351	.0417	.0905	-5.32	-36.59
216.00	.136	777.526	105.305	1040.432	656.336	.0407	.0921	-4.73	-37.27
240.00	-.154	779.264	108.543	1041.206	652.753	.0407	.0920	-4.81	-37.30
264.00	-.519	785.659	123.047	1044.047	700.294	.0418	.0901	-5.58	-36.57
288.00	-1.218	802.022	168.134	1051.245	765.336	.0438	.0871	-6.94	-35.44
312.00	-2.449	837.440	230.290	1066.497	851.614	.0465	.0836	-8.78	-33.74
336.00	-1.399	903.533	322.133	1093.847	952.422	.0493	.0803	-10.84	-31.93

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POWER INPUT MODULE *STEADY STATE LUBRICATED ANALYSIS* 1450 SHP

T E M P E R A T U R E   M A P

TEMPERATURES ARE IN DEGREES CELSIUS. THE FIRST 58 TEMPERATURES ARE CALCULATED, THE OTHERS ARE KNOWN

STEADY STATE TEMPERATURE CALCULATION, FINAL RESULT AFTER 6 ITERATIONS

CALCULATED TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
1	93.103	2	94.718	3	80.748	4	115.494	5	117.314
6	113.694	7	106.140	8	102.353	9	107.245	10	106.375
11	95.107	12	101.927	13	78.735	14	82.319	15	86.455
16	84.951	17	86.153	18	88.457	19	79.555	20	84.626
21	94.332	22	89.034	23	87.388	24	90.050	25	85.253
26	94.509	27	88.393	28	87.313	29	86.495	30	79.623
31	81.989	32	87.257	33	95.328	34	88.343	35	65.504
36	85.144	37	103.551	38	84.396	39	83.695	40	92.442
41	101.703	42	72.266	43	78.785	44	90.902	45	72.494
46	84.750	47	94.713	48	72.632	49	85.078	50	95.235
51	73.183	52	86.022	53	95.677	54	95.415	55	93.130
56	93.922	57	101.778	58	87.757				

KNOWN BOUNDARY TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
59	75.000	60	24.000	61	70.000				

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